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DESIGN PERFORMANCE AND OPERATIONAL CHARACTERISTICS OF THE ARL TWENTY-INCH HYPERSONIC WIND TUNNEL

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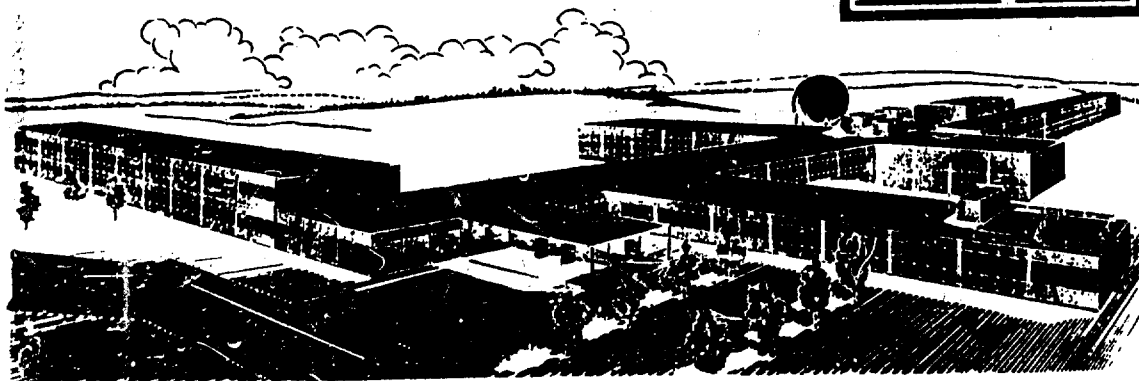THE OHIO STATE UNIVERSITY

COLUMBUS, OHIO

MARCH 1962
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FOREWORD

This interim technical report was prepared by The Ohio State University, Columbus, Ohio, on contract AF 33(616)-7451, for the Aeronautical Research Laboratories, Office of Aerospace Research, United States Air Force. The work reported herein was accomplished on Task 7065-01, "Fluid Dynamics Facilities Research" of Project 7065, "Aerospace Simulation Techniques Research". Technical monitors for the contractual period January 1960 to April 1962 were Mr. Fred Daum of the Hypersonic Research Laboratory and Mr. Emil Walk, Fluid Dynamics Research Facilities Laboratory of ARL.

ABSTRACT

The ARL Twenty-Inch Hypersonic Wind Tunnel is a blow-down type, free-jet installation that can deliver air at Mach numbers from 8 to 14 at free stream Reynolds numbers of 10^7 per foot to 0.3×10^6 per foot.

The report discusses some of the initial considerations influencing the configuration of the facility and presents theoretical performance estimates of its aerodynamic components; i.e., the electric resistance heater, the nozzle, the diffuser and the vacuum supply. Operational characteristics such as mass flow rates, electric power requirements, sphere evacuation rates, etc., are also treated for the full range of Mach numbers. Brief discussions on the extension of testing time and on the character of the delivered air flow are included.

Mechanical design, control systems, and wind tunnel instrumentation of the facility are not considered.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF SYMBOLS

c_p	Specific heat of air at constant pressure
c_v	Specific heat of air at constant volume
h	Enthalpy
h_c	Convective heat transfer coefficient, Eq (A-3)
k	Conductivity of air
m	Mass
$\dot{m}$	Mass flow rate
n_1	Number of first stage vacuum pumps
n_2	Number of second stage vacuum pumps
t	Time
u	Velocity
v	Volume of vacuum sphere
x	Distance from nozzle throat
A	Area
B	Vacuum pump constant, Eq (B-6)
C	Vacuum pump constant, Eq (B-6)
D	Wire Diameter
E	Internal energy
K_1	Constant for nozzle contour, Eq (6)
K_2	Constant for nozzle contour, Eq (6)
M	Mach number
P	Pressure
P_{t2}	Total pressure behind a normal shock
Q	Power
R	Gas constant for air
Re	Reynolds number

T	Temperature
V	Volume flow rate
W	Work
X	Nozzle contour abscissa
Y	Nozzle contour ordinate
Z	Potential energy
γ	Ratio of specific heats
δ	Boundary layer thickness
δ^*	Boundary layer displacement thickness
μ	Viscosity of air
η	Pressure recovery efficiency
ρ	Density of air

Subscripts

e	Conditions at nozzle exit
f	Conditions at film temperature Eq (A-5)
i	Conditions at initiation of process
o	Conditions in stagnation region
D	Conditions in diffuser
H	Conditions in heater

DESIGN PERFORMANCE AND OPERATIONAL
CHARACTERISTICS OF THE ARL TWENTY-INCH
HYPERSONIC WIND TUNNEL

I.   INTRODUCTION

The Twenty-inch Hypersonic Wind Tunnel (HWT) of the Aeronautical Research Labora-
tories is designed to produce high Mach number air flows at moderate Reynolds num-
bers.   It is the second of a series of facilities being developed at ARL to examine
the flow phenomena associated with hypersonic air streams.   The first of the series,
a three-inch diameter, continuous flow, hypersonic wind tunnel is operational and
has been reported in Reference 1.   A third facility, a 50-inch exit diameter, Mach
20, blow-down type wind tunnel is presently under construction.

The Twenty-inch HWT is an intermittent flow, free-jet facility that uses con-
toured, axisymmetric nozzles to produce high Mach number air flows.   A Chicago Pneu-
matic air compressor provides a maximum stagnation pressure capability of 2500 psia;
a 1600 KW electric resistance heater allows stagnation temperatures of $2800^{\circ}R$ to be
reached.   A vacuum pumping station consisting of 12, two-staged Allis-Chalmers vacu-
um pumps and a 55,000 ft^3 vacuum sphere maintain the required pressure ratio across
the nozzle during tests.   The electric power for the entire facility is supplied
from a 2000 KVA substation.

Figure 1 presents the designed range of operation of the Twenty-inch HWT in
terms of Mach number and Reynolds number.   At a given Mach number, the highest Rey-
nolds number is defined by a combination of maximum available stagnation pressure
and liquefaction effects; the lowest Reynolds number is determined from the minimum
stagnation pressure which will maintain isentropic flow through the nozzle at the
maximum stagnation temperature.   For this curve, the minimum stagnation pressure was
obtained by assuming that a total pressure behind a normal shock of 30 mm Hg was
necessary.   This Mach number-Reynolds number map illustrates the broad simulation
capability for the Twenty-inch HWT.

An analysis of the major components of the facility, that is, the air heater,
nozzle, diffuser, etc., is the prime concern of this report.   The design criteria
for each component is discussed and the theoretical performance estimates presented.
In order that this report may serve as a guide to the planning of research programs
in the facility, the operational characteristics, for example, mass flow rates,
power requirements and run times, are also presented.   The mechanical design, in-
strumentation and control systems are not treated.

Following a brief description of the facility and its operation, each major
system is discussed and the performance analyses made.   The operational character-
istics are then presented, followed by the Appendices which give the derivations of
some of the pertinent equations.

II. DESCRIPTION AND OPERATION OF THE WIND TUNNEL

GENERAL DESCRIPTION

The Twenty-inch HWT is classified as a blow-down, free-jet, hypersonic facility that uses axisymmetric, contoured nozzles of 20-inch exit diameter to develop high Mach number flow. To produce this high Mach number, several major components are required. These are:

    (1) a high pressure air supply

    (2) an air heater

    (3) a nozzle

    (4) a pressure recovery system

    (5) a vacuum supply

Suitable controls, performance monitors, and a test cabin where the models may be inserted into the stream are also required. The complete test stand is shown schematically in Figure 2.

To provide a high pressure supply, a Chicago Pneumatic Air Compressor inducts air, passes it through silica gel dryers to remove water vapor, and then stores the air in tanks with a volume of 1400 ft^3 at pressures up to 3000 psia until needed for tests.

The high expansion rates required to produce hypersonic flows can cause static air temperatures to fall below the liquefaction point; to prevent liquefaction, then, the air must be heated prior to entering the nozzle. The heater for this facility is an electric resistance type made of many elements of "Kanthal" A-1 wire, a ferro-chromium alloy, formed into loops supported on alumina rods. The total heater is divided into five sections, each "Y" connected section supplied from a three-phase 460-volt bus and separately controlled. Maximum design temperature capability of the heater is 2800°R and maximum power dissipation is near 1600 KW.

After heating, the air is accelerated to the desired level through the nozzle. The nozzle has an axisymmetric contour and is designed to use interchangeable throats. Two throats are presently available, one delivering a nominal M = 12 flow and the other delivering a nominal M = 14 flow. These throat sections are copper and use back-side water cooling to dissipate the high heat flux produced during tests.

Once the flow has passed through the test section the stream must be slowed with a minimum of losses. The deceleration is accomplished by a diffuser, which for this facility is axisymmetric and consists of a convergent section, a constant area section, and a diverging portion leading into a heat exchanger. The heat exchanger decreases the air temperature, protecting the vacuum pumps, which have an inlet air temperature limit of 130°F and decreasing the volume flow rate to the pumps. The heat exchanger is considered part of the pressure recovery system as a sizable pressure drop may occur across it, adversely influencing the over-all operating efficiency.

2

In order to maintain isentropic flow through hypersonic nozzles, large pressure ratios are necessary. To reduce the required stagnation pressure and still keep the flow isentropic, a vacuum may be applied to the downstream portion of the nozzle. For this facility, this vacuum is supplied by the 12 Allis Chalmers 27-D vacuum pumps and by the vacuum sphere.

The vacuum pumps have a two-stage configuration, 10 vacuum pumps exhausting into two. The maximum pumping rate is 25,000 ft^3/min.

OPERATION

When a test is to be made, the wind tunnel has the nozzle throat for the specified Mach number fitted in place. The desired Reynolds number is determined by a proper choice of stagnation pressure and stagnation temperature. These conditions also fix the power required of the air heater to produce the temperature rise and consideration can be given to the proper distribution of the power in the heater. When these advanced selections have been made, the test may proceed.

The first step is to evacuate the vacuum sphere using the 12 vacuum pumps. The sphere is then closed off and the pumps stopped. High-pressure air is started through the heater to a by-pass valve which directs the air to the atmosphere.

The by-pass valve is a requirement of the system because of the intermittent operation of the facility. A time interval of five to ten minutes is required to stabilize the air heater at the proper temperature, hence, a system to by-pass the vacuum sphere is necessary. Also, available electric power does not allow simultaneous operation of the vacuum pumps and air heater when large heating rates are required. The possibility of partial power distribution to the air heater and of partial power distribution to operate a portion of the vacuum pumps exists and will be discussed in a later section.

With air at the proper stagnation pressure flowing through the heater, variable resistors for the first four heater sections are positioned and the controllable last section set. The electrical power to the sections is then turned on in sequence. As the air temperature approaches the desired stagnation temperature, power to the last section is monitored and controlled automatically so that the stagnation conditions are reached and then maintained.

When the stagnation conditions are steady, the vacuum valve to the sphere is opened and the by-pass valve closed. The flow then expands through the nozzle into the test section and from there into the diffuser. Data are taken from the time that flow is established until the flow breaks down because of a decrease in the nozzle pressure ratio as the sphere pressure increases.

After the isentropic flow breaks down, the power is turned off and the by-pass valve opened to allow the heater to cool by exhausting air into the atmosphere. The tunnel vacuum valve is closed and the vacuum pumps started to evacuate the sphere to begin another test sequence. Depending upon the particular test conditions and the sphere pressure at the end of the test, the cycle time between tests may vary from 20 to 30 minutes.

During the test, several critical performance parameters are monitored to ensure safe operation of the facility. Besides the stagnation temperature and

3

stagnation pressure, which are maintained and recorded automatically, temperatures in key areas, such as the nozzle throat, heater wires, heat exchanger entrance, vacuum pump entrance, cooling water, etc., are recorded constantly and a panel alarm system is triggered by any over-temperature condition. The operator may then take proper action correcting the malfunction or terminating the test with emergency shut-down procedures.

III. DESIGN PERFORMANCE

The performance of the major systems of the Twenty-inch HWT will now be considered and theoretical performance curves presented. These analyses are extensions of the initial design calculations and include discussions of the design criteria influencing the particular component.

HIGH PRESSURE AIR SUPPLY

The high pressure station is part of the physical plant that services the three hypersonic facilities. Application of the station to the intermittent Twenty-inch HWT operation requires only that the compressor provide a sufficiently high storage pressure to allow the maximum design stagnation pressure of 2500 psia to be maintained for the duration of a test run.

For a typical test, assuming a mass flow rate of 3 slugs/minute for 10 minutes, the storage pressure would drop less than 200 psia. Therefore, if initially charged to 3000 psia, the 1400 ft^3 storage tanks provide an adequate supply for two tests at the maximum stagnation pressure of 2500 psia. Allowing for the cycle time required to evacuate the sphere between tests, during which the compressor can be operated to replenish the storage tank pressure according to the curve of Figure 3, at least one more test at maximum pressure may be performed.

Considering the down time in any research program, that is, the time required to alter or change models, calibrate instrumentation, etc., the limit of three consecutive tests at 2500 psia is not considered a disadvantage. Further, the majority of the tests will be conducted at stagnation pressures below maximum and at lower mass flow rates than the illustration; hence, the compressor station will serve the Twenty-inch HWT adequately.

The Kemp Silica-Gel dryers are also considered as part of the high pressure air supply. These dryers have the capability of delivering air at 3000 psia to the storage tanks at dew-points of -60°F; a sufficiently low value to assure dry air for test purposes.

AIR HEATER

The air heater is required to deliver heated air to the nozzle of the wind tunnel over a wide variety of operating conditions. To provide this air at temperatures close to the melting point of the heater material, a careful analysis of the heater is necessary. In the initial design phase, two criteria present themselves:

4

(1) The air heater must be able to dissipate enough power to heat the maximum design mass flow to the design temperature.

(2) The air heater's size must be compatible with the physical plant.

The maximum mass flow is determined by many factors: the over-all size of the facility, the Mach number range, the maximum pressure available, etc. For the present wind tunnel, the available physical plant resulted in a facility with a nozzle exit diameter of 20 inches, a Mach number range from 8 to 14, and a maximum pressure of 2500 psia. These factors fix the maximum mass flow rate. The temperature requirement is determined from the stagnation temperature necessary to maintain a liquefaction free air flow.

The electrical power required to obtain this temperature can be determined from the relation

$$Q = \dot{m}\ (h_{mx} - h_{in}) \times 1.356 \times 10^{-3}\ KW \tag{1}$$

where

$\dot{m}$ is mass flow rate in slugs/sec

h is enthalpy in ft lbs/slugs

The maximum temperature also determines the material with which the heater may be made. As this facility uses air as the working fluid, the choice of high temperature, easily worked, thermal shock resistant material that resists oxidation is limited. A ferro-chromium wire, Kanthal A-1, was chosen for use in the heater. Its melting temperature is $3210^{\circ}R$ and it is relatively easily worked.

With the total power known and the type of material selected, the configuration of the heater can be examined to determine the size. The method of power dissipation is mainly through convection from the resistance elements and therefore is governed by the expression

$$Q = h_c A\ (T_{wire} - T_{Air}) \tag{2}$$

where

A is the wire surface area and,

h_c is the convective heat transfer coefficient which for staggered circular cylinders is found to be

$$h_c = \left(\frac{k}{D}\right) 0.26\ Re^{.6}\ Pr^{.33} \tag{3}$$

These two equations lead to two important considerations. The first is the power density, Q/A, the power dissipated per unit area of wire surface. The second consideration is the velocity through the heater which must be sufficiently high to maintain the forced convection process.

For a fixed h_c in Eq (2), note that the higher the power density, the greater the temperature difference between wire and air. To allow air temperatures to increase to values close to the wire temperature for this design—an air temperature of 2800°R is required with a wire temperature limit of 2920°R for maximum continuous operation—a low power density is necessary.

However, when power density is low, a large amount of wire is required to absorb a given power. As this wire must be packaged in a configuration which will fit the physical size requirement, it is necessary to examine techniques to minimize the length of the heater. An obvious way to reduce the length is to increase the heater cross section; however, as the heater cross section increases, the velocity through the heater will decrease, adversely affecting the convective heat transfer.

From the equation governing the heat transfer coefficient, h_c, Eq (3), the two parameters which may be adjusted to maintain h_c at a high value are the wire diameter and the Reynolds number; the other factors are properties of the fluid. As the wire diameter is fixed by practical considerations, such as strength and workability at 0.144-inch diameter, (a standard size) only the Reynolds number may be varied, and this by changing the cross-sectional area of the heater and thus varying the velocity through the heater.

The air velocity through the heater may be found from the continuity relation

$$\dot{m} = \frac{P_o \, A^*}{60.5 \, \sqrt{T_o}} = \rho_H \, u_H \, A_H \tag{4}$$

after substitution of $P_H = \rho_H \, RT_H$ and assuming $P_H = P_o$

$$u_H = \frac{R}{60.5} \frac{A^*}{A_H} \sqrt{\frac{T_H}{T_o}} \tag{5}$$

The heater cross-section area A_H can be selected to provide the proper velocity through the heater. Two velocity limits must be considered:

    (1) A maximum velocity, determined by the dynamic pressure loads on the wire and supporting structure.

    (2) A minimum velocity, determined by the loss of the forced convection process.

Of the two limits, the second is the most critical. Experience has shown that, at velocities under two feet per second and near maximum temperatures, the free convective currents within the heater are of the same order as the velocity through the heater, giving an uneven vertical temperature distribution which can result in wire failure. With these factors considered for the required wide range of heater operation, the optimum cross section for the heater was chosen and is shown in Figure 4. The velocities through this heater are given for selected conditions in Figure 5. The minimum velocity refers to velocity through the open portion of the heater, the maximum velocity refers to the velocity past the wires, through the minimum area of the heater cross-section.

6

An effective method by which to achieve a significant reduction in heater length is to take advantage of an increased power density at the cold end of the heater. In this manner, the temperature difference between wire and air can be made large, and proper design can be used to maintain the wire temperature at safe levels. The power distribution for this heater is shown in Figure 6. Values of the power density vary from 188 watts/in^2 at the cold air inlet to 30 watts/in^2 at the hot air exit. These values are based on the hot resistance values; at 2800°R the wire resistance is 4% higher than at room temperature.

The final configuration of the heater can now be settled. An analysis outlined in Appendix A and presented in Figure 7 indicates the heater may deliver the design stagnation temperatures safely at M = 12 and M = 14. Temperature requirements for M = 10 and M = 8 are low enough and velocities high enough such that no critical conditions are reached.

Initial plans called for the first four sections (5, 4, 3, and 2) of the heater to be controlled by a simple "full on" or "all off" arrangement. The last section (I) was to be controllable and to adjust automatically the power required to maintain the proper temperature. For low power applications, sections were to be turned off as required. This type of operation is shown in Figure 7b. To increase the flexibility of operation, variable resistors were added to each section to give complete control over the operation of the heater. For low power applications it is now possible to use all the sections, but at a reduced power density, thereby giving a more favorable operation of the heater. This type of operation is shown in Figure 7c.

NOZZLE

An axisymmetric, contoured nozzle is fitted to the wind tunnel. The throat sections for the contour are interchangeable, enabling the contour to serve for several delivered Mach numbers by using throat sections of various diameters. This concept of a "Poly-Mach number" nozzle contour was advanced in Reference 2. While this contour is not "exact," in the sense of a contour obtained from the method of characteristics, its use in this facility was dictated by many factors:

(1) For each desired Mach number there is one contour which will be obtained from characteristic theory for a given initial divergence angle. Therefore, for each Mach number, a new nozzle is required.

(2) Present day knowledge of hypersonic growth rates in nozzles under high pressure and temperature gradients is limited, and, as the boundary layer growth must be added to the inviscid contour to obtain the physical configuration, any inaccuracies in the prediction of the boundary layer will result in an inviscid contour which is not the desired characteristic one.

(3) As boundary layer growth is a function of Reynolds number, the nozzle would be limited in its operating range of Reynolds number if it is to maintain a close approximation of the contour.

(4) The excessive length of the exact contour creates other problems for this facility, for example:

7

i   Installation of a long nozzle in a facility where
    there is a physical space limitation.

ii High nozzle starting pressure ratios, which appear
    to be functions of the nozzle throat to diffuser
    throat distance as reported in Reference 3.

iii Very large boundary layer thicknesses which decrease
    the effective testing diameter.

With these considerations in mind, a minimum length contour was determined
from an expression of the form

$$Y_e - Y = K_1 (X_e - X)^2 + K_2 (X_e - X)^5 \qquad (6)$$

The constants K_1 and K_2 are obtained by matching the contour to the end point and
slope of the diverging conical portion of the throat. This procedure is described
in Reference 2 to approximate the exact contours obtained by characteristic theory.

To this inviscid contour, the boundary layer thickness must be added to obtain
the physical contour. An empirical relation for the displacement thickness, found
to be valid for axisymmetric nozzles in this Mach number, Reynolds number range is,
Reference 2,

$$\frac{\delta^*}{x} = 0.0064 \frac{M^{1.25}}{(Re)^{.14}} \qquad (7)$$

where the Mach number and Reynolds number are based on the free stream conditions
at the nozzle exit.

The final contours are shown in Figure 8. The boundary layer displacement
thickness is based on a design Mach number of 14 and a free stream Reynolds number
of 10^6 per foot. The core of isentropic flow is also shown in the figure. It has
been estimated from the empirical relation

$$\frac{\delta}{\delta^*} = 2.5 \qquad (8)$$

Note that the edge of the boundary layer, δ, penetrates into the test rhombus
of uniform flow defined by a Mach wave emanating from the inviscid curve. This vis-
cous region decreases the test radius at the 190-inch station from 8.5 inches to
less than 3.5 inches. From the figure it appears that shortening the nozzle may
reduce the apparent region of uniform flow, but will increase the usable core. For
this reason, the nozzle contour was cut off at the 120-inch station, giving a usable
core of about 4.8 inches. The slight penalty of Mach number gradient and flow angu-
larity in the region above the Mach wave was accepted for the 37% reduction in length
of the nozzle.

8

For ease of manufacture, the nozzle is constructed of sections of steel tubing. A contour lathe follows a template matching the physical contour and machines two sections of tubing at a time. In this manner, the joints between each section are machined together keeping them smooth and maintaining a continuous contour. In general, the joints are not critical because of the thick boundary layer covering the nozzle; however, in the throat region, where the boundary layer is still thin, care must be used to keep the joints as smooth as possible to avoid undesirable wave patterns.

The throat sections are pure copper to make use of its high conductivity to transfer the heat from the throat to the back side water cooling tubes. Estimates of the heat transfer rate in the nozzle throat using the technique of Reference 4 indicate that the cooling is adequate to keep throat temperatures below $300^\circ F$.

PRESSURE RECOVERY SYSTEM

The deceleration process of the hypersonic stream after it has passed through the test section is quite important. Efficient pressure recovery can extend the running time and/or increase the Reynolds number range of the facility by enabling operation at low stagnation pressures. This pressure recovery is accomplished in a diffuser, which is not easily treated analytically. Consider that the hypersonic diffuser encloses a thick, cooled, boundary layer that is subjected to high adverse pressure gradients, shock impingements and wave expansions, and that the diffuser must operate satisfactorily with models of different sizes and orientation, which effect the wave patterns and hence the boundary layer in the diffuser, and the need for an empirical approach to diffuser design may be recognized.

Previous work with wind tunnels of configuration similar to the present one, References 3 and 5, indicate that satisfactory performance is obtained from an axisymmetric diffuser design consisting of a converging section, a constant area section and, a diverging section. Such diffusers have efficiencies, η_D, above 85% where

$$\eta_D = \frac{P_{t_D}}{P_{t_2}} \times 100 = \frac{\text{total pressure at diffuser exit} \times 100}{\text{total pressure at diffuser entrance}} \qquad (9)$$

In order to take advantage of this previous experience, a diffuser configuration of this type was chosen. To specify the diffuser, four dimensions must be fixed:

(1) The diameter of the constant area section

(2) The length of the constant area section

(3) The convergence angle for the entrance scoop

(4) The divergence angle for the subsonic diffusion

The most critical of these parameters is the diameter of the constant area section. This throat area must be large enough to swallow the starting shock system -- allowing for the boundary layer and the presence of a model -- yet not

9

so large that once the diffuser has started that it will operate inefficiently. Rejecting the complexity of a variable geometry diffuser, and relying on the previous work a ratio of

$$\frac{\text{Diffuser throat area}}{\text{Nozzle exit area}} = .5625$$

was chosen for the facility.

The length of the constant area section is also of importance. The shock-down process does not occur immediately but, because of the shock wave-boundary layer interactions in the constant area section, occurs over an extended region. Length-to-diameter ratios for the constant area section of from 3:1 up to 5:1 have been used with success. Because of the space limitations of the facility, a ratio of 3:1 was selected.

The angles of the entrance scoop and diverging portion of the diffuser are not as critical as the previous two parameters. However, the entrance scoop does play an important part in the entrainment capabilities of the diffuser upon the free jet. As the optimum angle of the scoop depends upon a variety of conditions, Mach number and Reynolds number, as well as the model location and orientation as discussed in Reference 6, a compromise scoop configuration was selected. Figure 9 presents the final design of the diffuser.

For the purposes of discussion and calculation in the report, the heat exchanger will be considered a part of the pressure recovery system. The pressure downstream of the heat exchanger influences the volume flow rate capabilities of the vacuum pumps and the run time of the wind tunnel; therefore, the recovery efficiency η will be used to describe the total diffusion process where

$$\eta = \frac{P_H}{P_{t_2}} \times 100 = \frac{\text{total pressure downstream of heat exchanger}}{\text{total pressure at entrance of diffuser}} \times 100 \qquad (10)$$

The pressure downstream of the heat exchanger is made up of the recovered pressure from the diffuser, minus the pressure drop due to the heat exchanger. As the heat exchanger consists of five banks of water-cooled finned tubes transverse to the flow, sizeable pressure drops may occur across it, and recovery efficiency may be strongly effected by the heat exchanger.

VACUUM SUPPLY

The low pressure required to operate the facility is obtained from 12 Allis Chalmers type 27-D vacuum pumps. These pumps are used to evacuate the 35,000 ft^3 sphere to a low pressure prior to each test run of the wind tunnel.

In order to perform an analytic study of the pumping capacity of the facility its pumping characteristics must be known. An empirically derived relation for the volume flow rate, V, of one of these pumps is

$$V = 3150 - 157 \frac{P_2}{P_1} \quad \text{in ft}^3/\text{min} \tag{11}$$

where

P_1 is inlet pressure to pump

P_2 is outlet pressure from pump

This expression is used for all subsequent calculations with the vacuum pumps.

The vacuum pumping station uses a two-stage operation, ten pumps exhausting into two pumps. The advantages of staging are:

(1) lower sphere pressures

(2) higher volume flow rates at low inlet pressure

To illustrate the advantages of this staging, consider the following pumping configuration:

Let n_1 pumps be in the first stage

n_2 pumps be in the second stage

P_1 be inlet pressure to first stage

P_2 be interstage pressure

P_a be exit pressure of the second stage, the ambient pressure

For the first stage then, the volume flow rate is

$$V_1 = n_1 \left[3150 - 157 \frac{P_2}{P_1} \right] \tag{12}$$

For the second stage

$$V_2 = n_2 \left[3150 - 157 \frac{P_a}{P_2} \right] \tag{13}$$

Employing the continuity relation, and after substituting for the perfect gas relation and noting that a second stage heat exchanger will maintain a constant temperature:

$$V_2 = V_1 \frac{P_1}{P_2} = n_2 \left[3150 - 157 \frac{P_a}{P_2} \right] \tag{14}$$

11

Eliminating P_2 from Eq (12) and substituting into Eq (14) the volume flow rate into the first stage of a two-stage configuration is found to be

$$V_1 = \frac{n_2 \, (63,200 - 157 \, P_a/P_1)}{(1 + 20.05 \, \frac{n_2}{n_1})} \tag{15}$$

As a comparison, a single stage system of $n_1 + n_2$ pumps has a flow rate of

$$V = (n_1 + n_2) \left[3150 - 157 \, \frac{P_a}{P_1} \right] \tag{16}$$

The minimum pressure which can be reached occurs at $V = 0$; therefore for the two-stage system

$$P_1 = \frac{157}{63,200} \, P_a = 1.9 \text{ mm Hg}$$

and for the single stage

$$P_1 = \frac{157}{3150} \, P_a = 38 \text{ mm Hg}$$

The second effect of staging is shown clearly in Figure 10. The dashed line indicates six pumps in parallel and may be compared with the two-stage configuration of five into one vacuum pump. Staging has straightened the flow rate curve to some extent, keeping the flow rate almost constant at 12,500 ft³/min until an inlet pressure of about 20 mm Hg. Note that below 105 mm Hg it is preferable to use staging; however, above this pressure, paralleling the pumps will give higher pumping capacity. As low sphere pressure is required for this facility the staging system was used. Figure 10 presents the volume flow rate for several pumping combinations.

IV. OPERATIONAL CHARACTERISTICS

MACH NUMBER RANGE

This portion of the report deals with quantities of an operational nature, i.e., mass flow rates, power requirements, evacuation rates of the sphere, etc. To cover the entire range of operation of the facility, Mach numbers of 8, 10, 12 and 14 are considered. The nozzle throat sizes, based on the boundary layer growth rate of Eq (7) are:

Mach No.	Throat Size
8	1.312 inch
10	0.557 inch
12	0.408 inch
14	0.230 inch

12

MASS FLOW RATE

The mass flow rate delivered by the wind tunnel is an important characteristic which determines the power required for given stagnation conditions and the run time for the blow-down facility. In Figure 11, the mass flow rate is presented as a function of the stagnation pressure and stagnation temperature. Based on the relation

$$\dot{m} = \frac{P_o A^x}{60.5\sqrt{T_o}} \quad \text{slugs/sec} \tag{17}$$

the mass flow rate may be noted to vary from values of 2 slugs/sec at $M = 8$ to 0.02 slugs at $M = 14$.

POWER REQUIREMENTS

With the mass flow rate known for given stagnation conditions, the power required to obtain these conditions may be determined from the expression

$$Q = \dot{m} (h_o - h_1) \times 1.356 \times 10^{-3} \text{ KW.} \tag{18}$$

The influence of the large mass flow rates at $M = 8$ is clearly shown in Figure 12. The constant power curves indicate high temperatures possible only at low pressures; whereas at $M = 14$, the constant power curves blanket the entire pressure, temperature range.

EVACUATION RATE OF SPHERE

Prior to each test, the sphere must be evacuated. Depending upon the initial Mach number and the duration of the test, the required initial pressure can vary from 10 mm Hg to the minimum possible, 2 mm Hg. The rate of evacuation of the sphere is therefore seen to be an important parameter in the cycle time between tests. Following the derivation of Appendix B, the evacuation rate is found to be

$$P = \frac{B}{C} + \left[P_1 - \frac{B}{C}\right] e^{-\frac{Ct}{V}} \tag{19}$$

where B and C are constants that depend upon the number of vacuum pumps operating in each stage:

$$B = \frac{(152 \; n_2 \; P_a)}{60 \; (1 + 20.05 \; \frac{n_2}{n_1})}$$

(20)

$$C = \frac{63,200 \; n_2}{60 \; (1 + 20.05 \; \frac{n_2}{n_1})}$$

Figure 13a indicates the time required to reduce the sphere pressure from one atmosphere to below 100 mm Hg as for several vacuum pumping combinations. Figure 13b presents the rate for pressures below 100 mm Hg. The test run will be terminated when the sphere pressure has risen to a value such that the pressure ratio across the nozzle is insufficient to maintain isentropic flow. Hence, it will rarely be necessary to evacuate the sphere from atmospheric pressure, usually only from pressures of the order of 50 mm Hg. Using an 8 into 2 staging, then, the time required to pull sphere to 2 mm Hg from this level may be found from Figure 13b to be about 8 minutes.

TEST TIMES

As this facility is a blow-down type, the sphere pressure will increase as the tunnel mass flow rate discharges into it. For each operating stagnation pressure there is a maximum downstream pressure for which the flow in the nozzle will remain isentropic. When this downstream pressure is exceeded, flow will break down in the nozzle and the test will end. To determine the test time, therefore, two quantities must be known, the maximum pressure and the time it takes the sphere to reach this pressure.

Looking first at the rise of sphere pressure as a function of time and a constant delivered mass flow rate, Appendix C obtains the relation for time:

$$t = \frac{V}{\gamma \; \dot{m} \; RT_H} \left[P - P_1 \right] \quad \text{in seconds}$$

(21)

This expression, based on an adiabatic filling process, indicates that the pressure in the sphere varies linearly with time.

The pressure that the sphere will attain when the isentropic flow breaks down in the nozzle will depend upon the diffuser efficiency and pressure drop across the heat exchanger. Convention references diffuser efficiency to the normal shock pressure recovery based on nozzle exit Mach number; however, for convenience, the recovery efficiency defined in Eq (10) may be introduced. Making this substitution into Eq (21) along with the substitution for mass flow rate in terms of stagnation condition, Eq (17), the time to fill the sphere to drop out pressure becomes

14

$$t = \frac{60.5 \ v \ T_o}{\gamma \ A^* \ R \ T_H} \left[\eta \ \frac{P_{t2}}{P_o} - \frac{P_i}{P_o} \right] \text{ seconds} \qquad (22)$$

The duration of the test is then determined from the initial sphere pressure P_i, recovery efficiency η, stagnation conditions, T_o and P_o and through the normal shock total pressure ratio P_{t2}/P_o, Mach number.

Figure 14 presents run time as a function of these parameters for four Mach numbers. An initial sphere pressure of 2 mm Hg was assumed for the calculation and corrections for a thermally perfect gas were applied to the normal shock pressure ratio. The deviations from perfect gas with increasing temperature may be observed to decrease the allowable run times.

EXTENDED RUNNING

Although initial operation of the facility must be of the blow-down type described in the preceding section -- the power available is insufficient to operate the vacuum pumps, air heater, compressor and other electrical subsystems simultaneously -- the installation of an electric substation of larger capacity in the future warrants an examination of techniques to extend the running time of the facility.

Continuous operation of the facility depends upon the volume flow rate delivered by the tunnel to the vacuum pumping station. If the delivered rate can be accepted by the pumping station at a pump inlet pressure that is low enough to maintain an isentropic flow through the nozzle, the tunnel will operate continuously. The volume flow rate delivered by the tunnel V_H, may be obtained from the relation

$$V_H = \frac{\dot{m}}{\rho_H} = \frac{R A^* \ T_H}{60.5 \sqrt{T_o} \ \eta \ \frac{P_{+2}}{P_o}} \text{ in } ft^3/sec \qquad (23)$$

where

> T_H is the temperature of the heat exchanger, assumed constant, and where Eq (10) for recovery efficiency and Eq (17) for mass flow rate have been introduced.

The volume flow rates delivered at the four Mach numbers are given in Figure 15 as functions of stagnation temperature and recovery efficiency. These curves, together with the performance curves of the vacuum pumping station presented in Figure 10 indicated the conditions for which continuous running is possible. It may be observed that the high flow rates at Mach 8 and Mach 10 require a pumping rate much greater than the available capacity. At Mach numbers of 12 and 14, however, continuous operation does appear feasible for selected conditions.

Consider, for example, possible operation at Mach 12 with pump inlet pressure at 40 mm Hg. If 12 pumps are operated, Figure 10 shows that flow rates up to 24,000 ft^3/min may be handled. From Figure 15c the delivered flow rate for a given recovery efficiency is observed to decrease rapidly with stagnation temperature. Assuming $\eta = 70\%$, the delivered flow rate is 24,000 ft^3/min at 2200°R;

15

hence, at this temperature and under the above conditions, the tunnel would be able to operate continuously.

The recovery efficiency may be observed to influence the results in two ways: directly through the decrease in volume flow rate with increasing η, and indirectly through the increase in pump inlet pressure for a given stagnation pressure. It may be noted from Figure 10 that the vacuum pump volume flow rate increases with inlet pressure, until the "plateau" is reached.

Because starting efficiency is somewhat lower than running efficiency, it is possible that even with matched volume flow rates the tunnel will not go into flow. To alleviate this problem, it is possible to use the vacuum sphere to boost the available pressure ratio at the start of the test. This may be done by first evacuating the sphere to a low pressure, sealing it off, and then passing the flow through the tunnel into the vacuum pumps. When the required stagnation conditions are reached the sphere valve may be opened and the tunnel brought into flow. The sphere pressure will increase and, as the pressure rises the vacuum pumps will accept more of the delivered volume flow. If the vacuum pumps are able to handle all the delivered flow at an inlet pressure below drop-out pressure, the tunnel will run continuously. Operation of the pumps during the filling process will extend the running time by decreasing the flow rate into the sphere; to determine the increase in time, the following analysis is made.

The mass flow rate entering the sphere is the difference between that delivered by the tunnel and the portion accepted by the vacuum pumps, i.e.,

$$\dot{m} = \dot{m}_o - \dot{m}_p \quad \text{slugs/sec} \tag{24}$$

where

$\dot{m}_o$ is now the tunnel mass flow rate,

$\dot{m}_p$ is the mass flow rate accepted by the pumps.

As the pump volume flow rate is a function of inlet pressure, the mass flow accepted by the pumps will also vary with inlet pressure:

$$\dot{m}_p = \rho v = \frac{CP - B}{RT} \tag{25}$$

Making this substitution into Eq (24) and then solving for the pressure rise of the sphere with time, one obtains

$$t = \frac{v}{\gamma R T_H} \frac{(P - P_i)}{\left[m_o + \frac{1}{RT} (B - C \bar{r}) \right]} \tag{26}$$

In this expression the temperature depends upon the pressure.

16

Recalling that we have assumed an adiabatic filling process, the relation between temperature and pressure may be expressed as

$$T = \gamma \, T_H \left[\cfrac{P}{P - P_i \left(1 - \gamma \, \cfrac{T_H}{T_i} \right)} \right] \tag{27}$$

which is noted to have a maximum of $\gamma \, T_H$ as initial pressure goes to zero. Figure 16 presents the sphere temperature rise.

The time to reach a given pressure can now be determined as a function of tunnel mass flow rate. This is shown in Figure 17 where the mass flow rate is presented in slugs per minute, and the initial conditions are a sphere pressure of 2 mm Hg and sphere temperature of $540^\circ R$. The dashed lines of the curve indicate the pressure vs. time relation of Eq (21), when all the tunnel flow is ducted to the sphere; the solid curves indicate pressure – time relations when all the vacuum pumps are operating.

If the mass flow rate is 2 slugs/minute, Figure 17 shows that operating the pumps during the filling process will extend the time to 30 mm Hg sphere pressure from slightly more than a minute to more than seven minutes. Note also that at the low mass flow rate of 1 slug per minute, the tunnel will be able to run continuously if the drop-out pressure is greater than 16 mm Hg. Based on this curve, then, sizeable extensions of run time may be expected, provided that power is available to operate all the vacuum pumps.

It is fortunate that the criterion for continuous or extended running time, low mass flow rates, also lowers the required heater power, thereby increasing the possibility of extending the duration of the tests. The precise evaluation of the conditions required to extend run times, however, must await actual operation of the facility.

STREAM PROPERTIES

Because of the high Mach number, moderate stagnation temperature operation of this facility, some comments are in order concerning the properties of the air stream.

To obtain the stream properties based on an isentropic expansion from the stagnation conditions, the perfect gas ratios must be modified to account for the excitation of the vibrational energy made above stagnation temperatures of $1000^\circ R$. Corrections to the perfect gas ratios may be obtained from Reference 7. These corrections have been used, when applicable, to all the calculations in this report.

In Figure 18, the free stream Reynolds number per foot is presented. Besides corrections for a thermally perfect gas being required to obtain free stream density, the viscosity must also be altered from the usual Sutherland's viscosity relation. The Sutherland's viscosity law holds for temperatures above $180^\circ R$; as the stream static temperatures can be below $100^\circ R$, a more appropriate viscosity relation is that of Bromley-Wilke, Reference 8. This relation is presented in Figure 19 and

17

has been used in the Reynolds number calculations of Figure 18 when static temperatures were less than 180°R. Above this temperature, the Bromley-Wilke's and Sutherland's viscosity expressions coincide.

In order to assess the possibility of air liquefaction in the air at low temperature, the vapor pressure of air must be known for the pressure and temperature ranges obtained in the stream. No experimental data exists at this time for the vapor pressure of air in these ranges; therefore, the pure oxygen and pure nitrogen vapor curves of Reference 9 are given in Figure 20.

Observations of the allowable degrees of supersaturation in hypersonic streams vary from 0° to 30°R depending upon the particular facility and experiment. The reasons for the large discrepancies are variously attributed to condensation of nuclei upon water vapor, carbon dioxide, dust particules and to measurement techniques (i.e., the inability to sense the onset of condensation). At this stage, it appears that while some amount of supersaturation can be supported, the actual value for this facility must await the results of the initial test program.

V. SUMMARY

The Twenty-inch HWT of the Aeronautical Research Laboratories is a complex installation that requires close examination of the design criteria in order to obtain the maximum performance from its many components. Based on the initial specifications of a facility to operate intermittently over a Mach number range from 8 to 14, and to be as large as possible in order that relatively large size models could be tested, the present configuration evolved.

The intermittent operation required a vacuum sphere to provide the proper pressure ratio across the nozzle, but allowed a relatively small (compared to delivered volume flow rates at M = 8, for example) vacuum pumping station to be used to evacuate the sphere. Blow-down operation also fitted in with the requirement of separate operation of the vacuum pumps and air heater because of insufficient available electric power for simultaneous operation. However, a potential problem area, the mechanical design of a by-pass system with a valve that must operate freely against pressures of 2500 psia and temperatures of 2800°R was introduced by this intermittent operation.

Besides the obvious increases in mass flow rates and power requirements with size of the facility, direct influences upon the design of the air heater, the nozzle, and the diffuser are felt due to size alone. In order to fit within the laboratory area:

(1) The air heater must be designed with a high power density to minimize its length.

(2) The contoured nozzle must be as short as possible (this also has the advantage of keeping boundary layer at a minimum).

(3) The diffuser must have the minimum length of constant area section.

18

These design compromises have been examined and result in the configuration of the facility that has been considered.

In addition to the presented performance data for the various systems, the operational characteristics of the Twenty-inch HWT are given. The mass flow rates and power requirements are well defined once the throat area is fixed; however, several characteristics cannot be established until actual operation of the wind tunnel. The curves for run time and volume flow rate are given as a function of the recovery efficiency η, which is unknown at this time; although the strong influence of η upon run time, volume flow rate, and hence, possible continuous operation, can be observed.

With regard to continuous running, estimates have been made of the effect of operation of the vacuum pumps during the blow-down period. For low mass flow rates, it appears that the vacuum pump operation can result in extended run times and possibly, continuous running of the facility. The requirement of low mass flow rate also means that low power is necessary, hence, even without an increase in the available electric power, extended runs may be possible.

Other characteristics of the facility have been examined. For example, the evacuation rate of the sphere for various pump combinations, Reynolds number range capabilities and stream properties have been presented in the appropriate curves.

Liquefaction effects cannot be resolved until operation of the facility begins. It is apparent from previous work that some amount of supersaturation of the air components can be tolerated, but the exact amount appears to differ with facilities. If large degrees of supersaturation can be obtained in the facility, the Reynolds number operating range may be extended, and the heater life increased by operating at lower temperatures.

This report, then, has considered the basic design and performance of the Twenty-inch HWT at ARL. By presenting the performance of the components and the characteristics of operation of the wind tunnel, this report can serve as a guide for the planning of research programs in the facility.

19

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APPENDIX

A. AIR HEATER ANALYSIS

In the previous discussion of the air heater, consideration was given to the design criteria which led to a heater of a given configuration. After a series of calculations which determined the proper power distribution, wire surface area, and heater cross-section required to provide a safe heater design, the final configuration of the heater was set. The method used for the calculation is the subject of this Appendix.

Consider a heater made up of many identical elements connected in series. For this heater the following is assumed known:

(1) Total power to be dissipated

(2) Diameter of the wire used in the elements

(3) Length of the wire used in the elements

(4) The cross-sectional area of the heater box

(5) The temperature of the entering air

The problem is to heat a given mass flow rate of air to the design temperature without exceeding the maximum wire temperature recommended for continuous running.

For a given mass flow rate, the total power required to heat the air to the desired temperature is determined from

$$Q_t = \dot{m} \left(h_{out} - h_{in} \right) \tag{A-1}$$

With this total power fixed, the power supplied to each element can be obtained. As the wire surface area that is available to dissipate the power is known, the wire temperature of the first element can be found from the expression

$$Q' = h_c A \left(T_{wire} - T_{air} \right) \tag{A-2}$$

where

Q is the power applied to the first element

T_{air} is the air temperature after absorbing the applied power (from Eq (A-1))

h_c is the convective heat transfer coefficient

A is the wire surface area

21

The convective heat transfer coefficient, h_c, can be estimated from the empirical relations governing fluid flow over a bank of staggered cylinders, from any heat transfer text,

$$h_c = 0.26 \frac{k_f}{D} Re^{.6} Pr^{.33} \qquad (A-3)$$

For convenience, Eq (A-3) may be rewritten in terms of the fluid properties C_p, μ, and k, and the mass flow rate $\dot{m}$:

$$h_c = 0.26 \frac{k_f}{D} \left(\frac{\dot{m} D}{\mu_f A_H} \right)^{0.6} \left(\frac{\mu_f C_{p_f}}{k_f} \right)^{0.33} \qquad (A-4)$$

where:

D is the diameter of the wire used in the element,

A_H is the minimum area of the heater box.

Subscript f refers to the evaluation of the fluid properties at the film temperature T_f, defined as

$$T_f = \frac{T_{wire} + T_{air}}{2} \qquad (A-5)$$

Note that the definition of the film temperature prior to solution of Eq (A-2) for the wire temperature implies an iteration procedure. This procedure does not complicate the analysis.

The above equations give the required information about the first element, i.e., the air temperature after heating by an amount Q and the wire temperature required to transmit the power Q to the air by convection. Each element may be handled in like manner; however, when the power density, Q/A, is constant, it is sufficient to examine only the first and last elements of the heater.

For the last element, the air temperature at the exit is given by A-1; therefore, applying Eq (A-2) with the proper h_c provides the wire temperature. As long as this wire temperature is less than the maximum allowable, the heater will be safe. If the wire temperature exceeds the allowable temperature, changes in the heater must be made, for example, a decrease in the power density. Equation (A-2) indicates that a decrease in power density will lower the difference between the wire and air temperature. Various combinations of the design parameters may be tried until the proper conditions are found.

The above analysis applies to a heater of constant power density. It has been observed that when the maximum heated air temperature is approached, the required wire temperature can exceed the allowable temperature unless the power density is

low. As a low power density, which requires much wire surface area to dissipate the total power, results in a heater of excessive length, it is advantageous to develop a heater with a varying power density.

The design procedure for a heater of varying power density follows along similar lines. Each section of constant power density may be treated by the above procedure, the exit condition for the first section being used as the entrance condition for the second section and so on. In this manner, electric power is transmitted into the air stream until the total power, defined by Eq (A-1), is used to heat the stream to the desired level. To allow control over the entire range of temperatures, at least one of the sections must have a fully variable power supply, the other sections may be simply designed to produce a constant power.

EVACUATION RATE OF SPHERE

To determine the evacuation rate of the sphere using the Allis Chalmers vacuum pumps, the following analysis, based on an isothermal process is used.

The mass of the sphere of volume v at any time is given by

$$m = \rho v \text{ slugs} \tag{B-1}$$

For an isothermal process the time rate of change of this mass

$$\dot{m} = \frac{v}{RT} \frac{dP}{dt} \text{ slugs/sec} \tag{B-2}$$

The vacuum pumps can evacuate a given mass of the air in the sphere, depending upon the volume flow rate V of the vacuum pumps:

$$\dot{m} = \rho V = \frac{P}{RT} V \text{ slugs/sec} \tag{B-3}$$

The vacuum pumps, however, have a flow rate dependent upon the pump inlet pressure, of the form

$$V = C - \frac{B}{P} \text{ ft}^3/\text{sec} \tag{B-4}$$

Where the constants B and C can be found from the general expression for the Allis Chalmers two stage pumping configuration:

$$V = \frac{n_2 \left(63,200 - 157 \frac{P_a}{P}\right)}{60 \left[1 + 20.05 \frac{n_2}{n_1}\right]} \text{ ft}^3/\text{sec} \tag{B-5}$$

23

Hence

$$B = \frac{157\ n_2\ P_a}{60\left[1 + 20.05\ \dfrac{n_2}{n_1}\right]} \qquad\qquad (B-6)$$

$$C = \frac{63{,}200\ n_2}{60\left[1 + 20.05\ \dfrac{n_2}{n_1}\right]}$$

Equating Eq (B-2) and Eq (B-3) after substitution of Eq (B-4) gives the differential equation:

$$\frac{dP}{P\left(C - \dfrac{B}{P}\right)} = \frac{1}{v}\ dt \qquad\qquad (B-7)$$

With the condition that $t = 0$ when $P = P_1$, the initial pressure, the solution of Eq (B-7) is

$$P = \frac{B}{C} + \left(P_1 - \frac{B}{C}\right)\ e^{-\frac{Ct}{v}} \qquad\qquad (B-8)$$

This general expression may be used to determine the evacuation rate of the sphere for any number of vacuum pumps in the first and second stages.

CALCULATION OF RUN TIME

In order to be able to compute the time available for testing, the rate of pressure increase in the sphere with time must be known. The derivation of the time rate of change of the sphere pressure for the case where the constant tunnel mass flow rate exhausts into the sphere is now considered.

The first law of Thermodynamics may be written

$$\delta Q = \delta E - \delta W \qquad\qquad (C-1)$$

where

δQ is the heat transferred to sphere

δE is the internal energy change within sphere

δW is the work done by the added mass

24

Considering the sphere filling process as an adiabatic process,

$$\delta Q = 0,$$

hence

$$\delta E = \delta W \qquad \text{(C-2)}$$

With constant specific heats, the left side of Eq (C-2) becomes

$$\delta E = E - E_i = m c_v T - m_i c_v T_i \qquad \text{(C-3)}$$

where the subscript i refers to the initial sphere condition. As the mass in the sphere can be written as

$$m = \rho v = \frac{P}{RT} v \quad \text{and} \quad m_i = \frac{P_i}{RT_i} v \qquad \text{(C-4)}$$

Equation (C-3) becomes

$$E - E_i = \frac{c_v \, v}{R} \left[P - P_i \right] \qquad \text{(C-5)}$$

The work done by the added mass can be expressed as

$$\delta W = \left[h + \frac{u^2}{2g} + Z \right] \delta m \qquad \text{(C-6)}$$

where

        h is the enthalpy

        $\frac{u^2}{2g}$ the kinetic energy

        Z the potential energy

        δm the added mass = $\dot{m} t$

The changes in kinetic and potential energy are negligible, therefore the work may be expressed as

$$\delta W = h \dot{m} t = c_p T_H \dot{m} t \qquad \text{(C-7)}$$

where the enthalpy has been evaluated at the temperature downstream of the heat exchanger T_H. Equating Eq (C-5) and Eq (C-7):

$$\frac{c_v V}{R} \left[P - P_i \right] = c_p T_H \dot{m} t \qquad (C-8)$$

Therefore

$$t = \frac{V}{\gamma \dot{m} RT_H} \left[P - P_i \right] \quad \text{seconds} \qquad (C-9)$$

The pressure is observed to increase linearly with time. The duration of the test will depend upon the sphere pressure which will support isentropic flow through the nozzle. Expressing this terminating pressure as

$$P = \eta \frac{P_{t_2}}{P_o} P_o \qquad (C-10)$$

where η is the recovery efficiency and P_{t_2}/P_o is the normal shock total pressure ratio, Eq (C-9) becomes the time to arrive at drop-out pressure.

$$t = \frac{V P_o}{\gamma \dot{m} RT_H} \left[\eta \frac{P_{t_2}}{P_o} - \frac{P_i}{P_o} \right] \qquad (C-11)$$
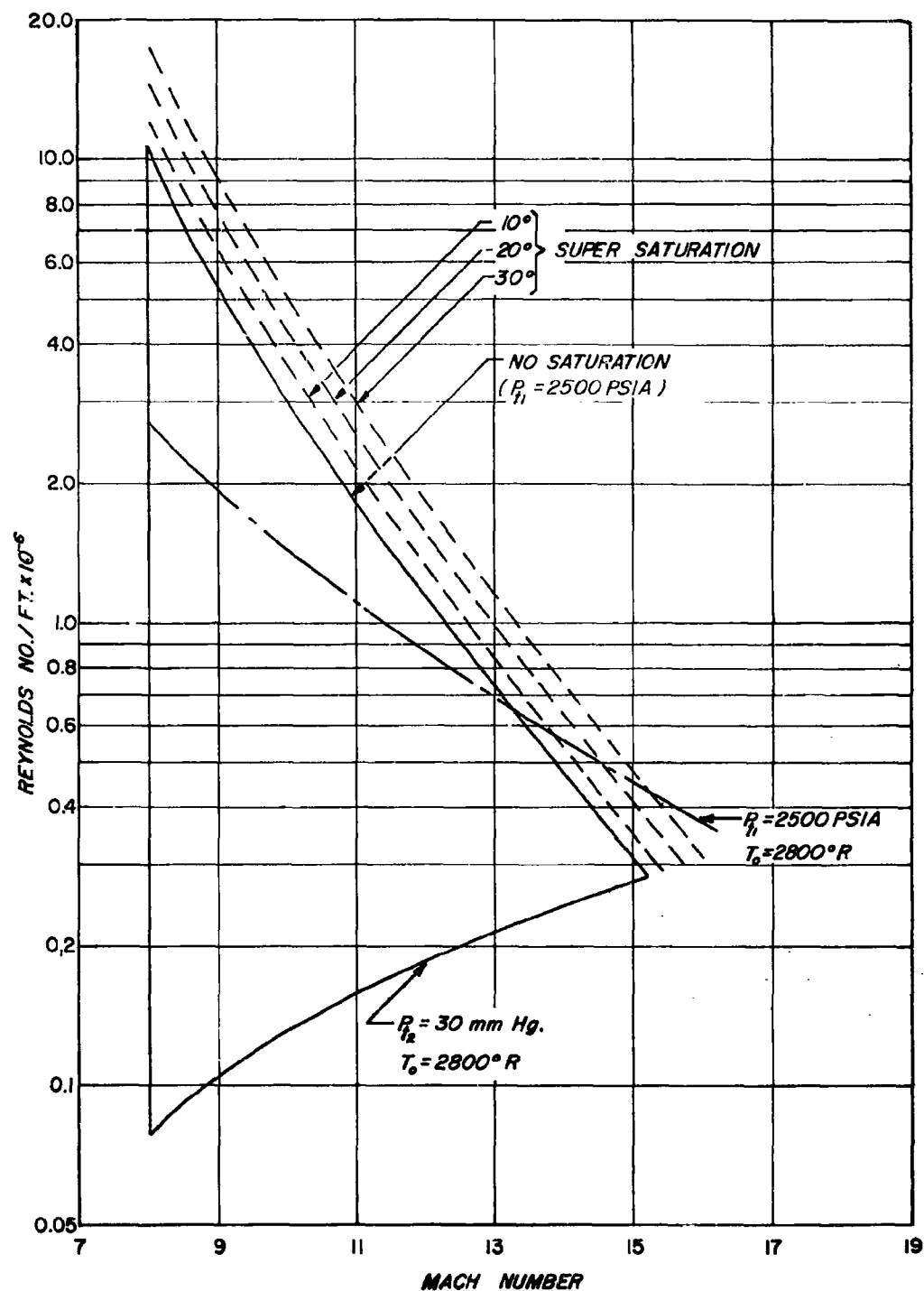
FIG. I   REYNOLDS NUMBER / FOOT vs. MACH NUMBER,
FOR OPERATING RANGE OF ARL 20 IN. HYPERSONIC WIND TUNNEL.

1 – STAGNATION PRESSURE CONTROL VALVE.
2 – HOT BYPASS VALVE.
3 – BYPASS TO ATMOSPHERE.
4 – TUNNEL VACUUM VALVE.
5 – PUMP HOUSE VACUUM VALVE.
6 – MAIN AIR SUPPLY VALVE.
7 – SPHERE VACUUM VALVE.
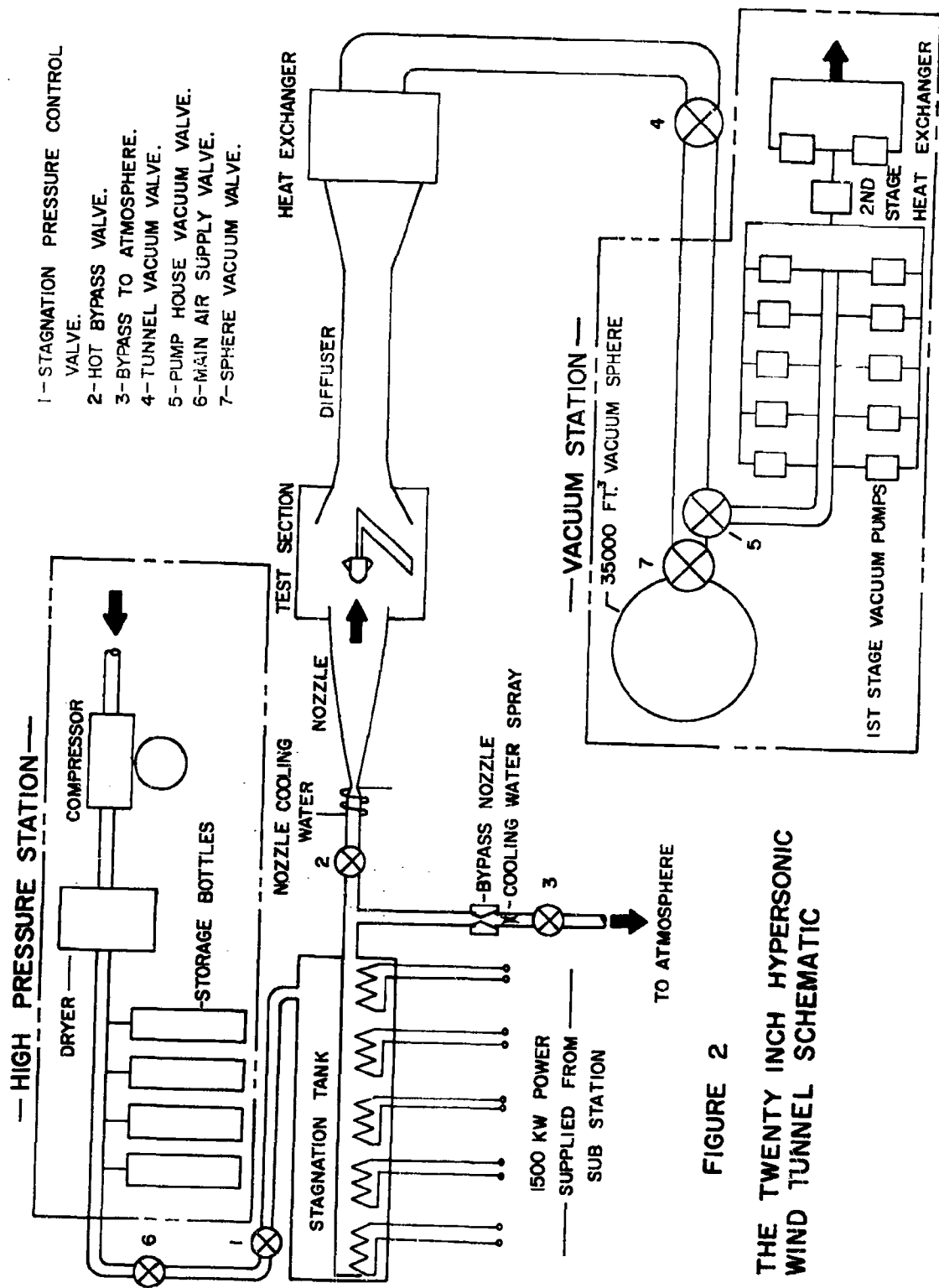
FIGURE 2

THE TWENTY INCH HYPERSONIC
WIND TUNNEL SCHEMATIC
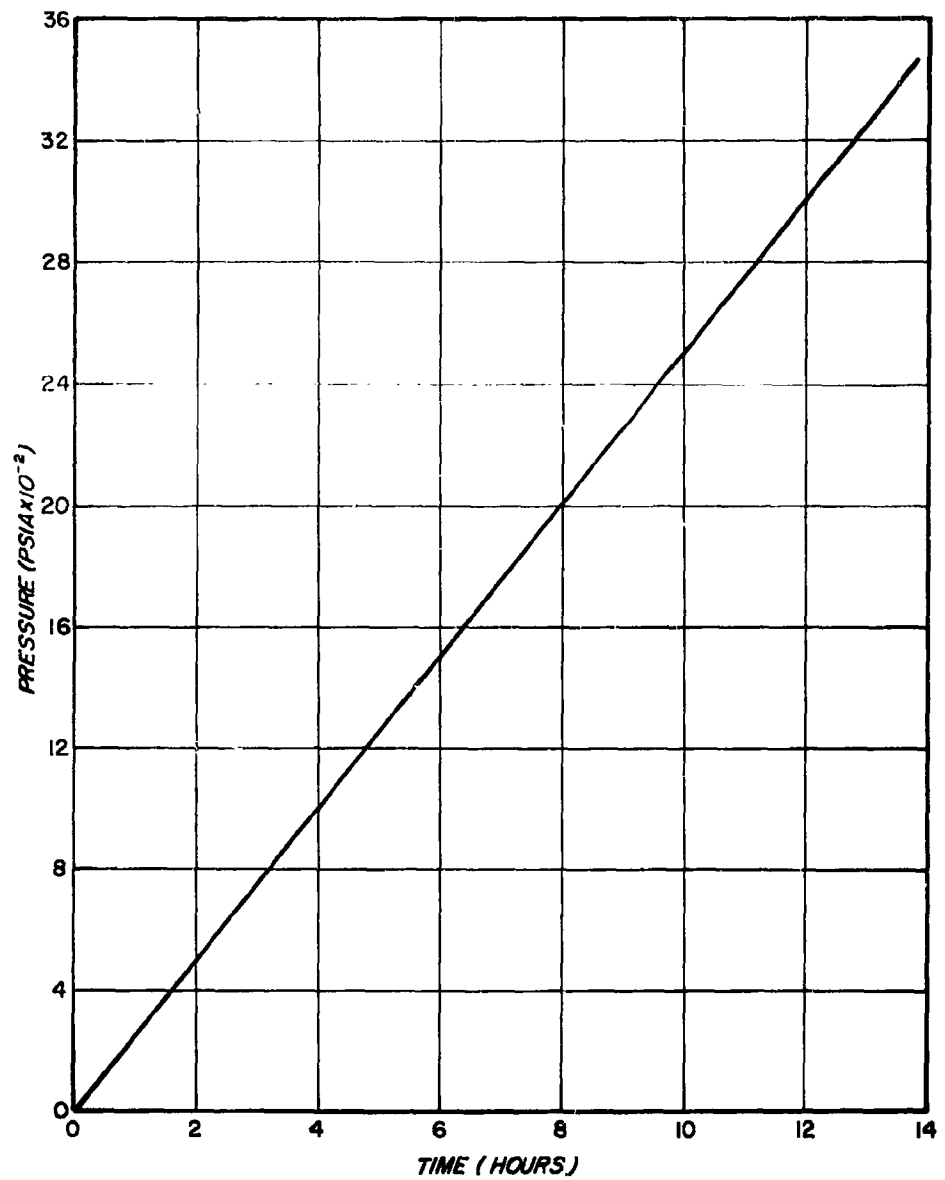
FIG.3   RESERVOIR PRESSURE *vs.* TIME FOR NORMAL
COMPRESSOR OPERATION.

29

FIG. 4 HEATER CROSS-SECTION

ALUMINA SUPPORT ROD

STEEL HEATER BOX

FIBERFRAX INSULATION

POWER FEED THRU

TYPICAL KANTHAL A-1 WIRE CONFIGURATION

PORCELAIN BUSHING

7"

5.625"

4.625"

5.50"

2.062"

30
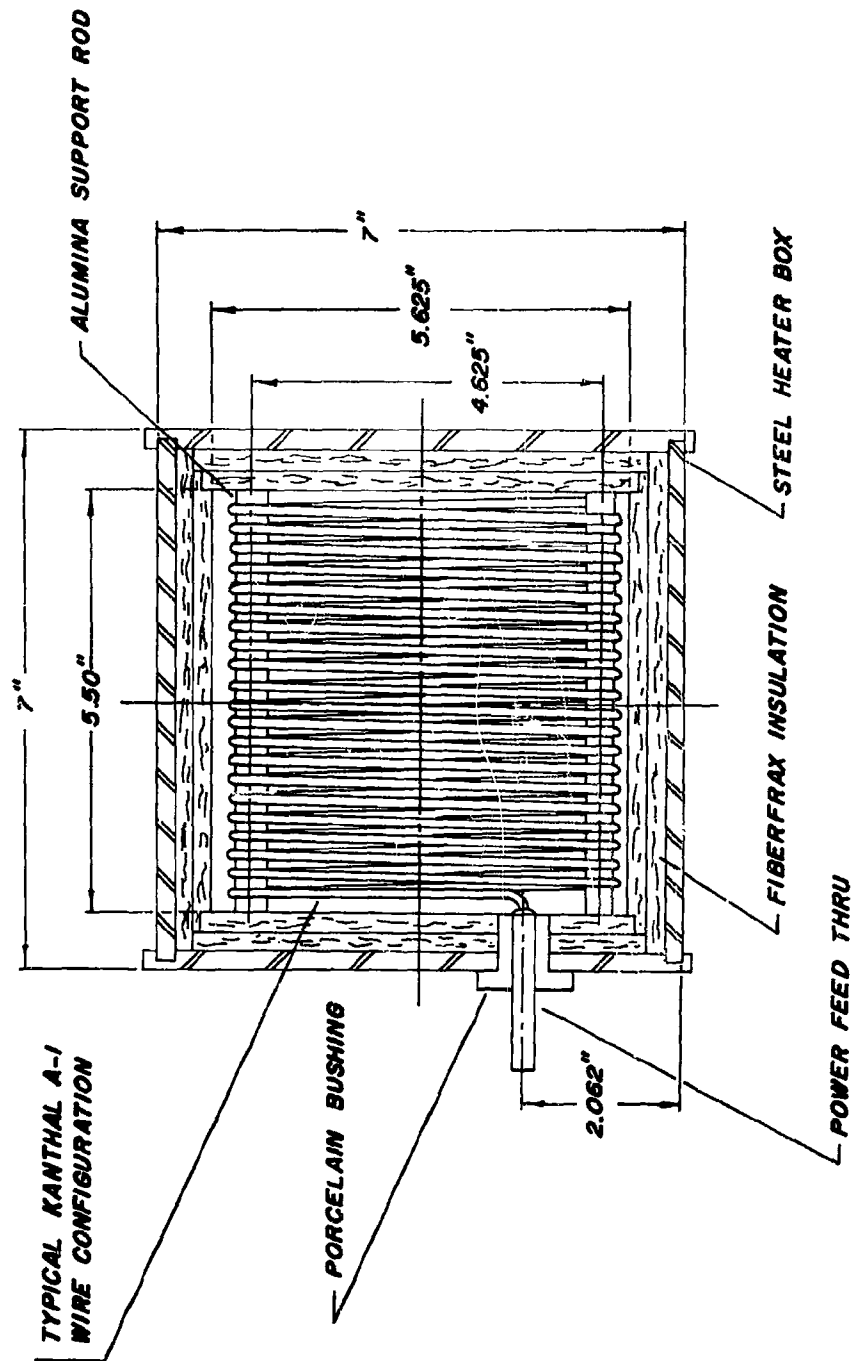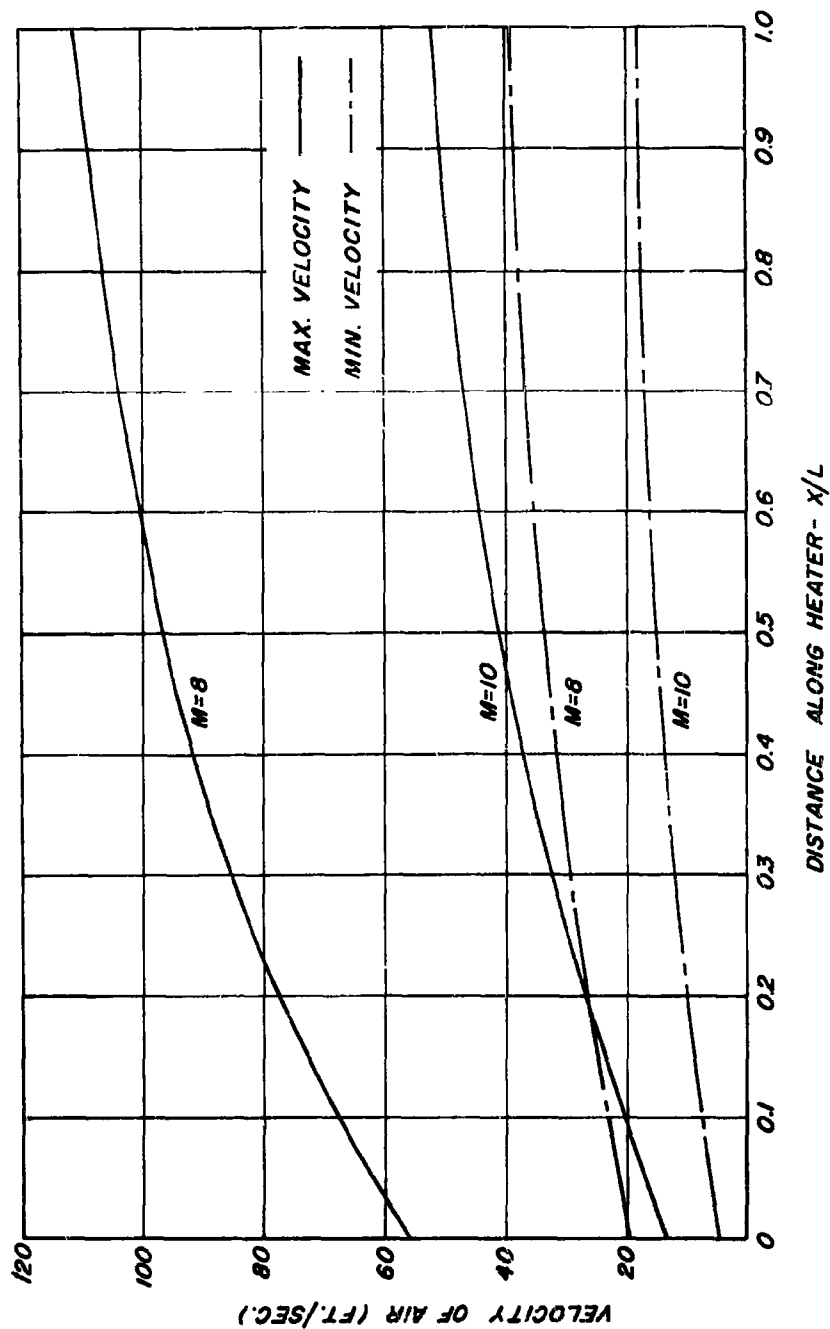
FIG.5a VELOCITY OF AIR THROUGH HEATER - M=8,10

DISTANCE ALONG HEATER- X/L

VELOCITY OF AIR (FT./SEC.)

MAX. VELOCITY ———
MIN. VELOCITY —·—·—

M=8

M=10

M=8

M=10
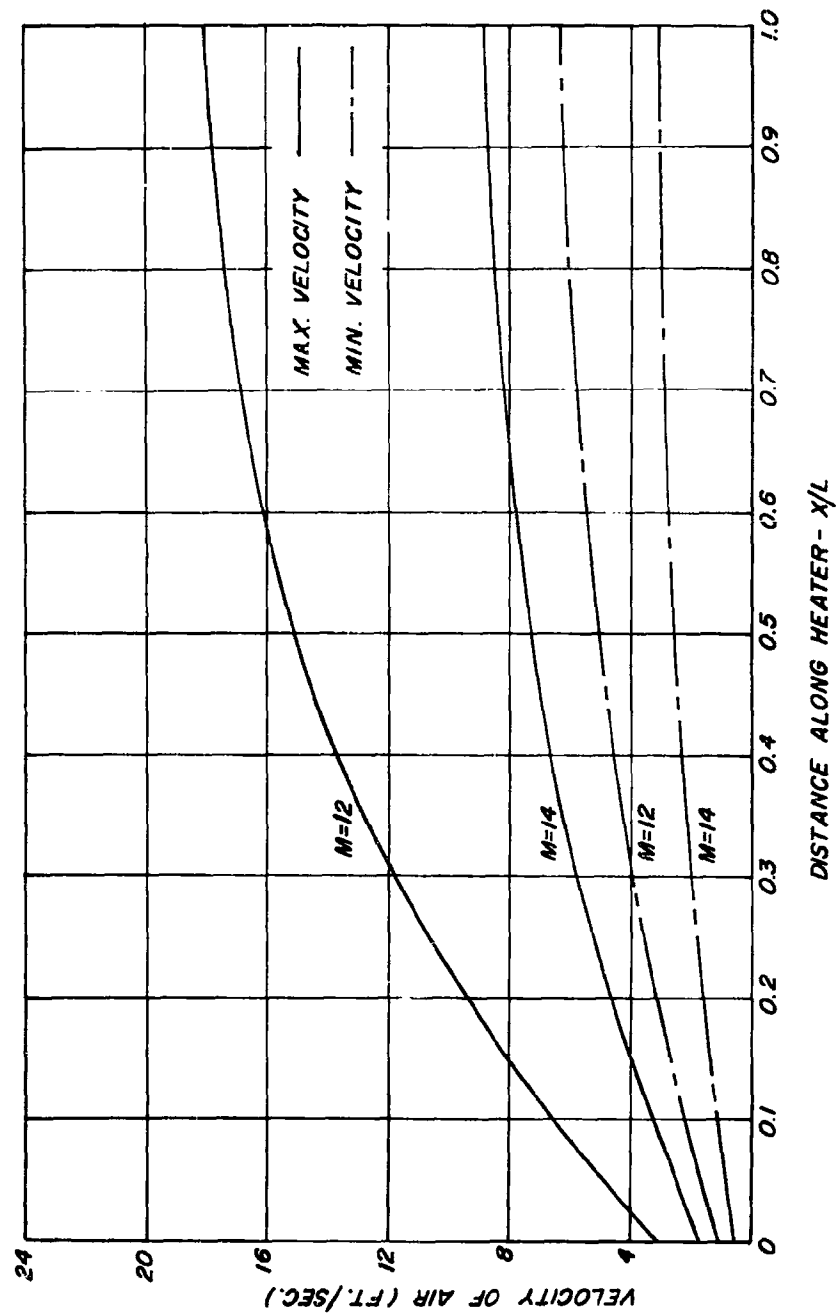
FIG. 5b VELOCITY OF AIR THROUGH HEATER - M = 12,14

HTR.NO.	WIRE SURF.AREA ft²	POWER kw
5	12.87	348.48
4	8.58	232.32
3	8.58	232.32
2	25.74	309.72
1 sec. 1	12.87	154.86
1 sec. 2	17.16	116.16
1 sec.3	10.725	46.47
TOTAL	96.53	1440.33
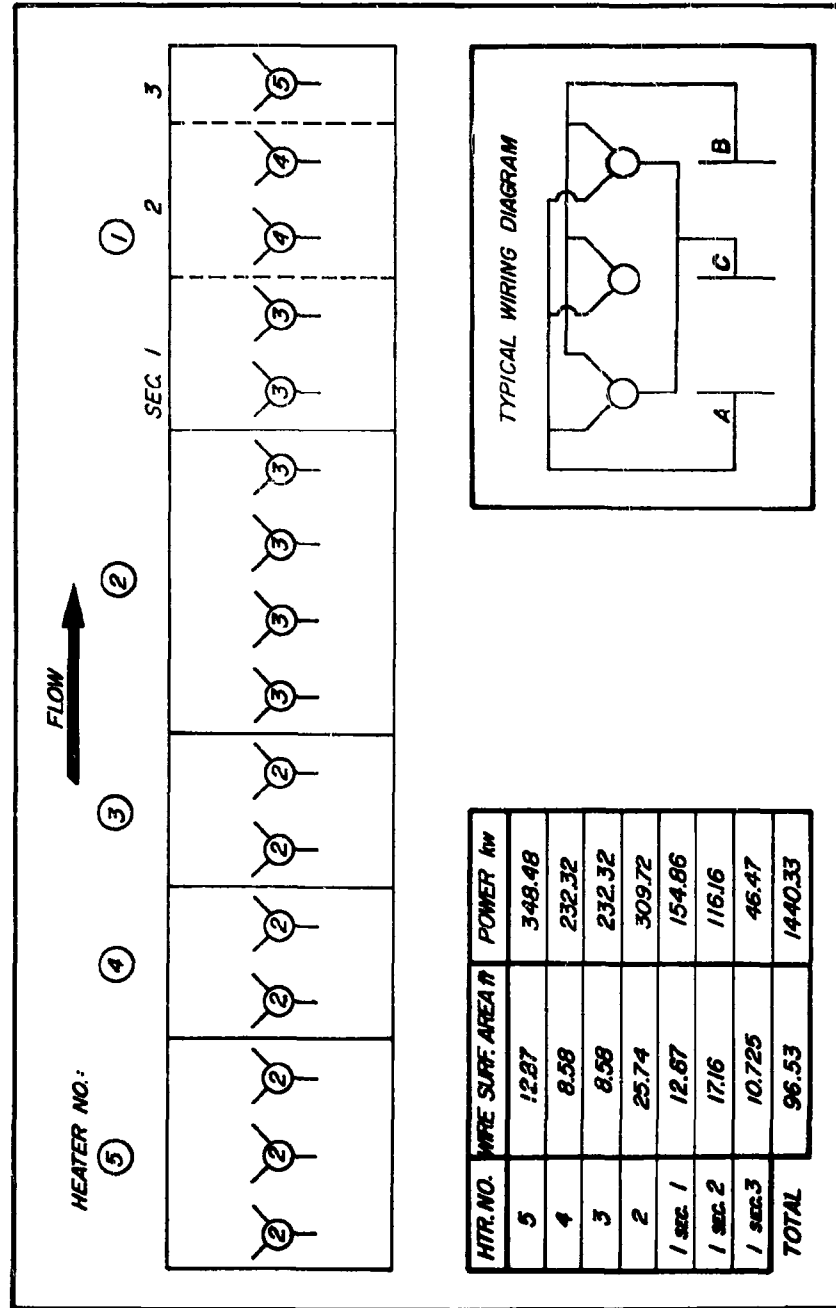
FIG. 6   SCHEMATIC OF ELECTRIC HEATER

33
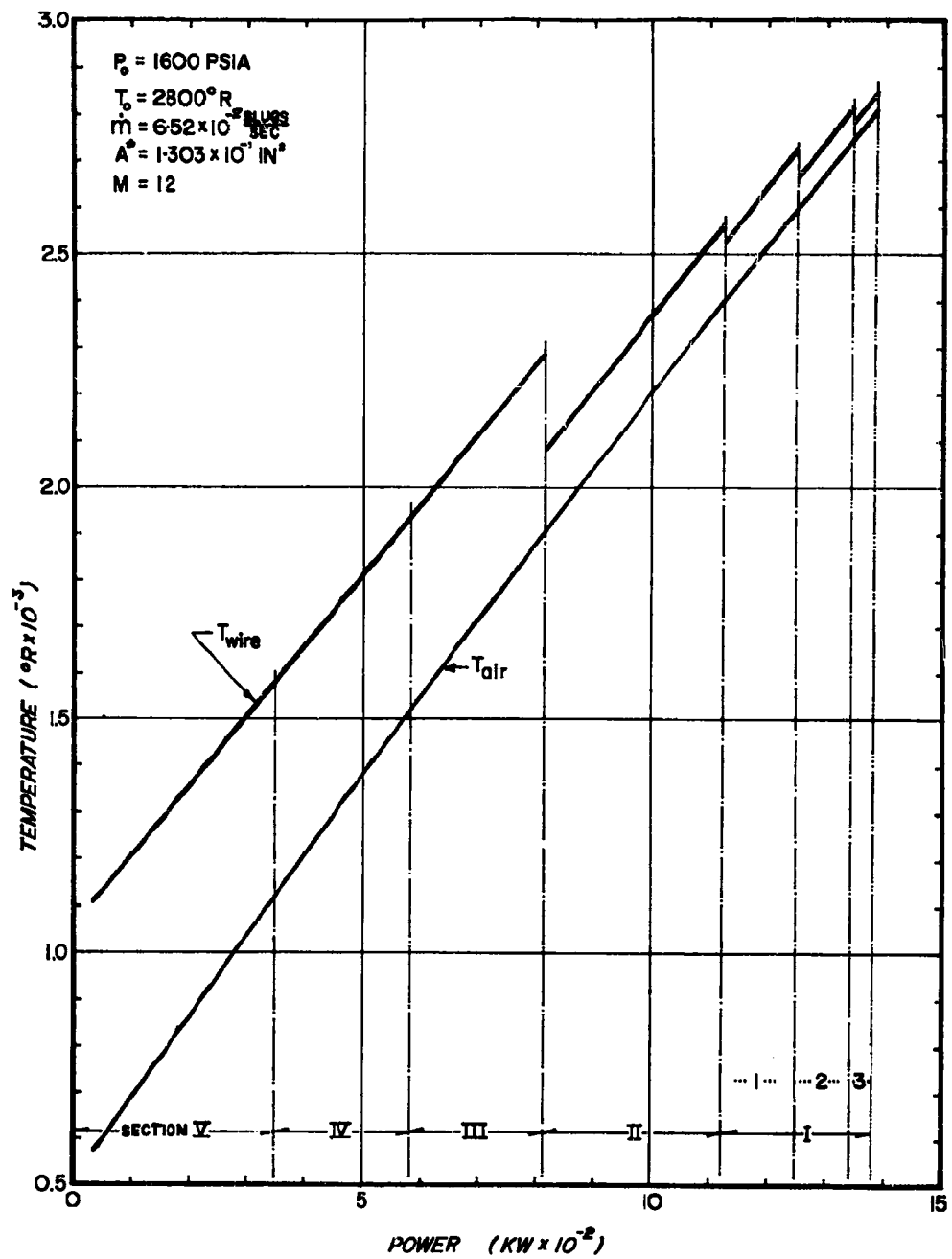
FIG. 7a AIR & WIRE TEMPERATURE *vs.* POWER INPUT.

34
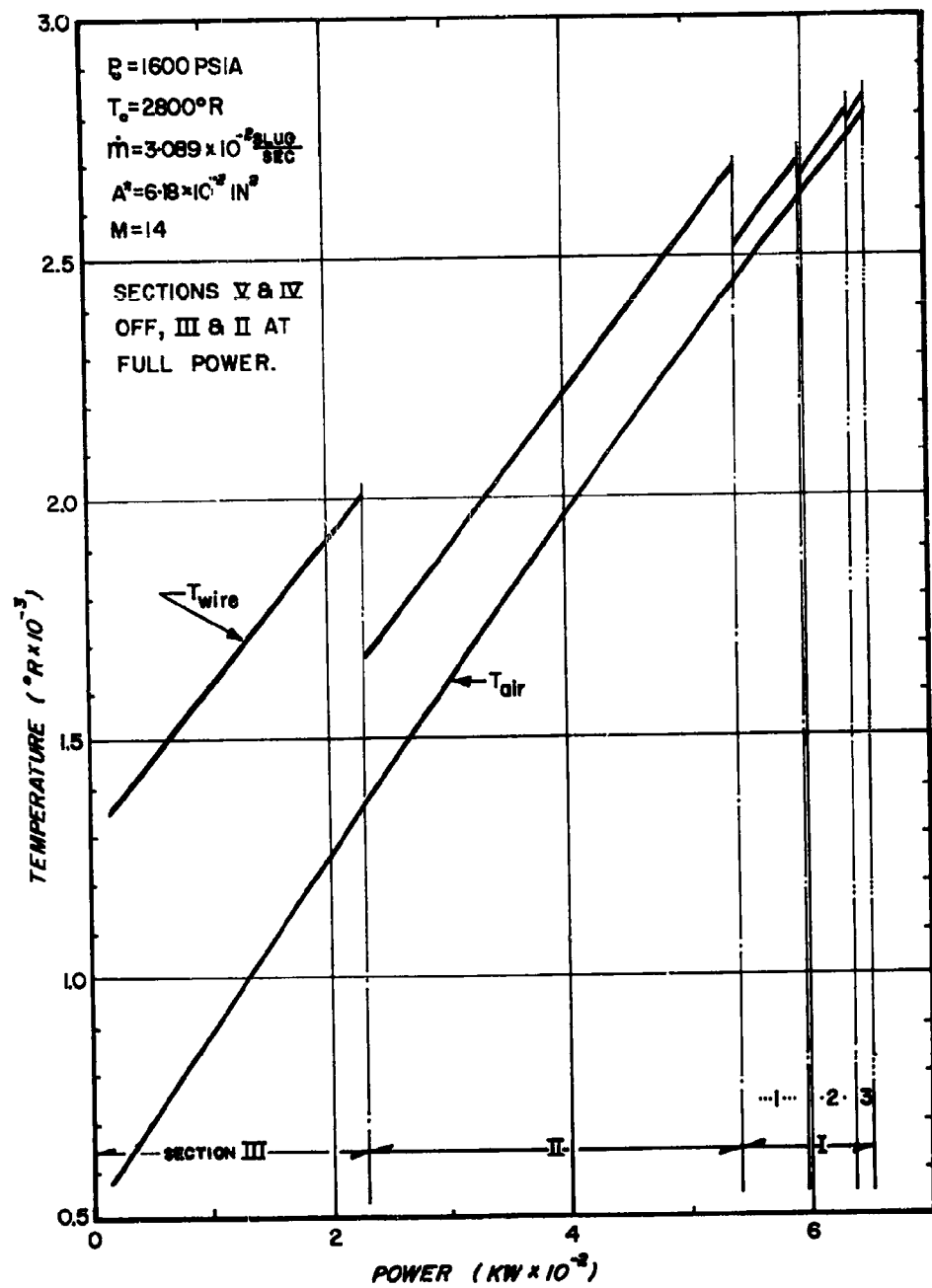
FIG. 7b AIR & WIRE TEMPERATURE vs. POWER INPUT.
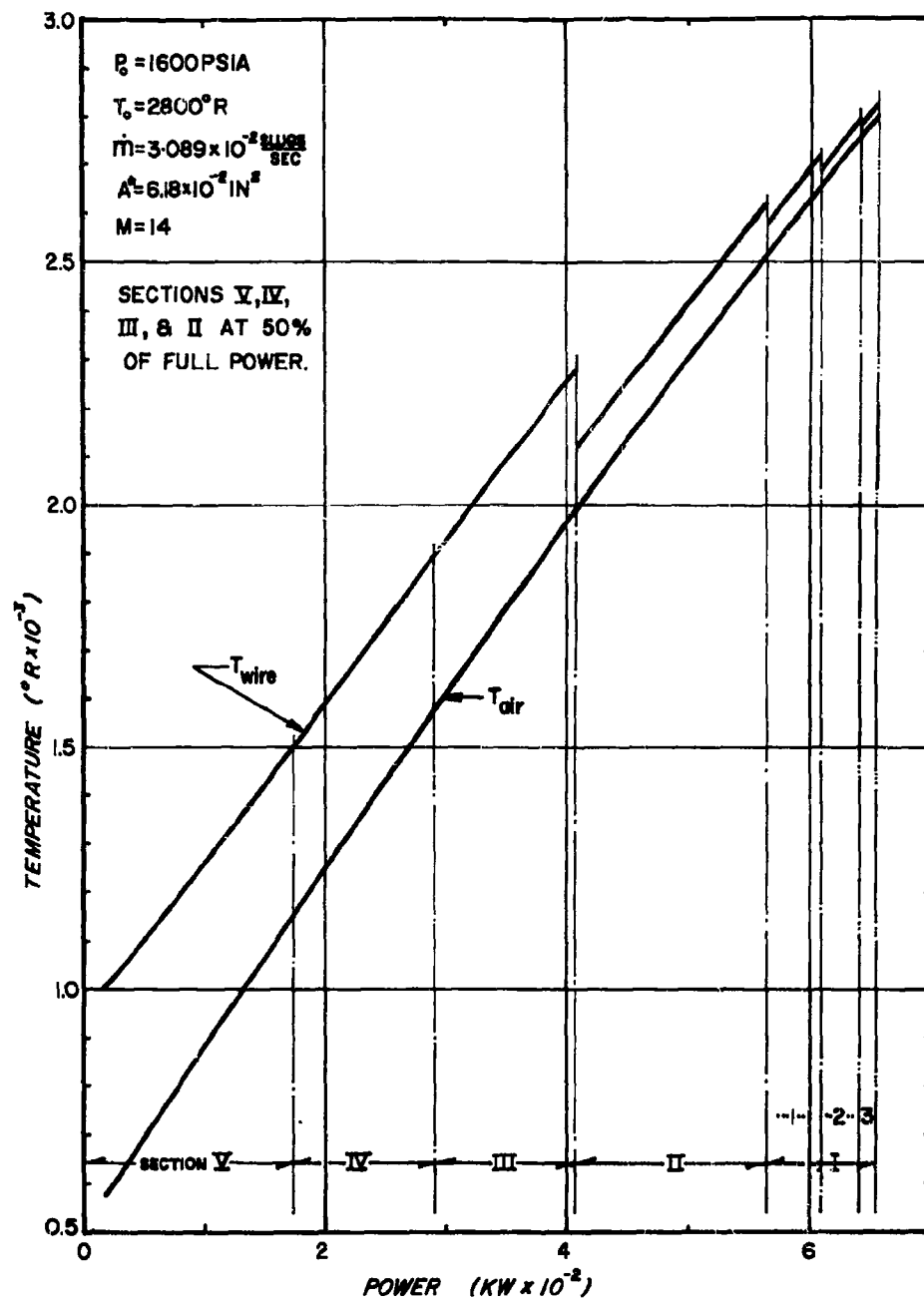
FIG. 7c AIR & WIRE TEMPERATURE vs. POWER INPUT.
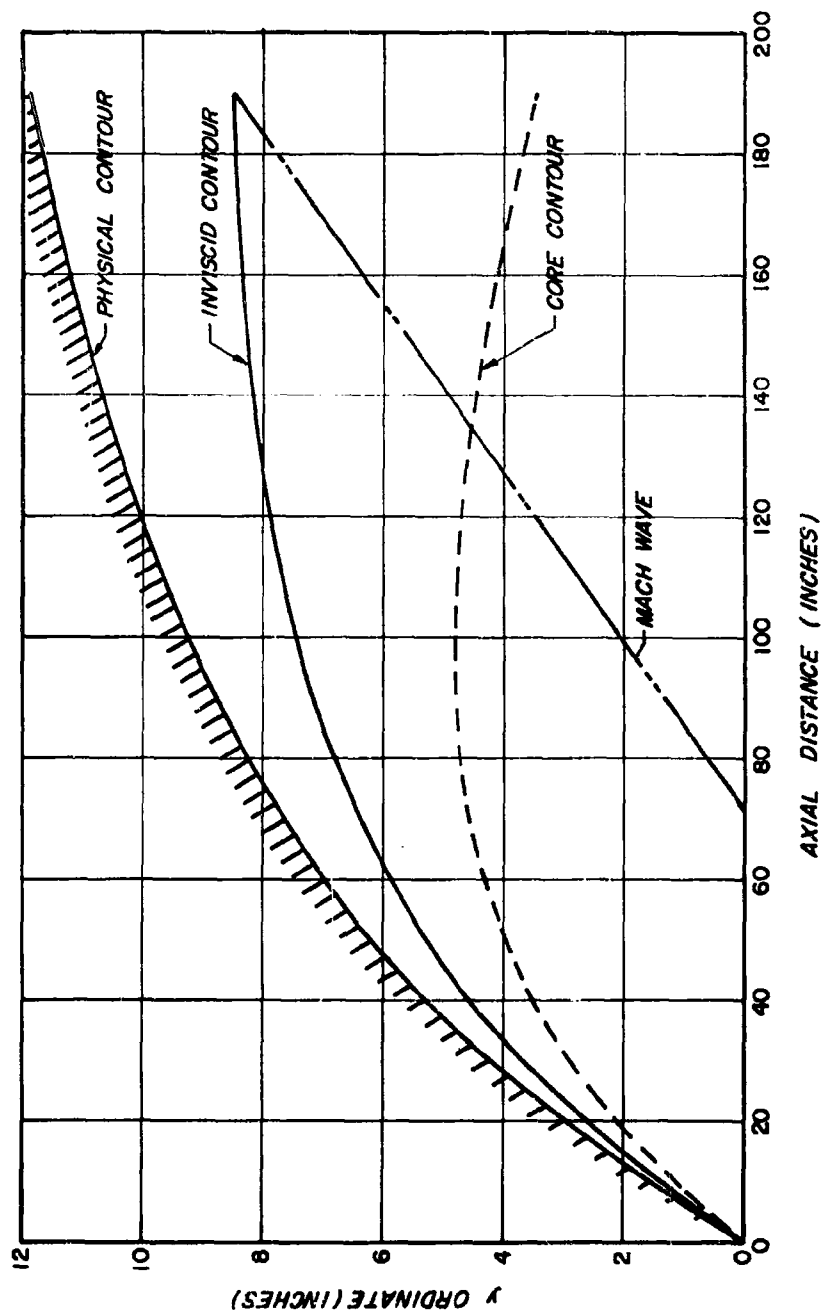
FIG. 8   CONTOURS FOR A MACH 14 NOZZLE CONFIGURATION.

37

DIFFUSER THROAT AREA = 0.5625
NOZZLE EXIT AREA

FLOW

30" I.D.
20" I.D.
15" I.D.
20" I.D.

HEAT EXCHANGER

33"

18.625"
16.75"
45"
35.75"
53.5"

14'-1.625" (REF.)
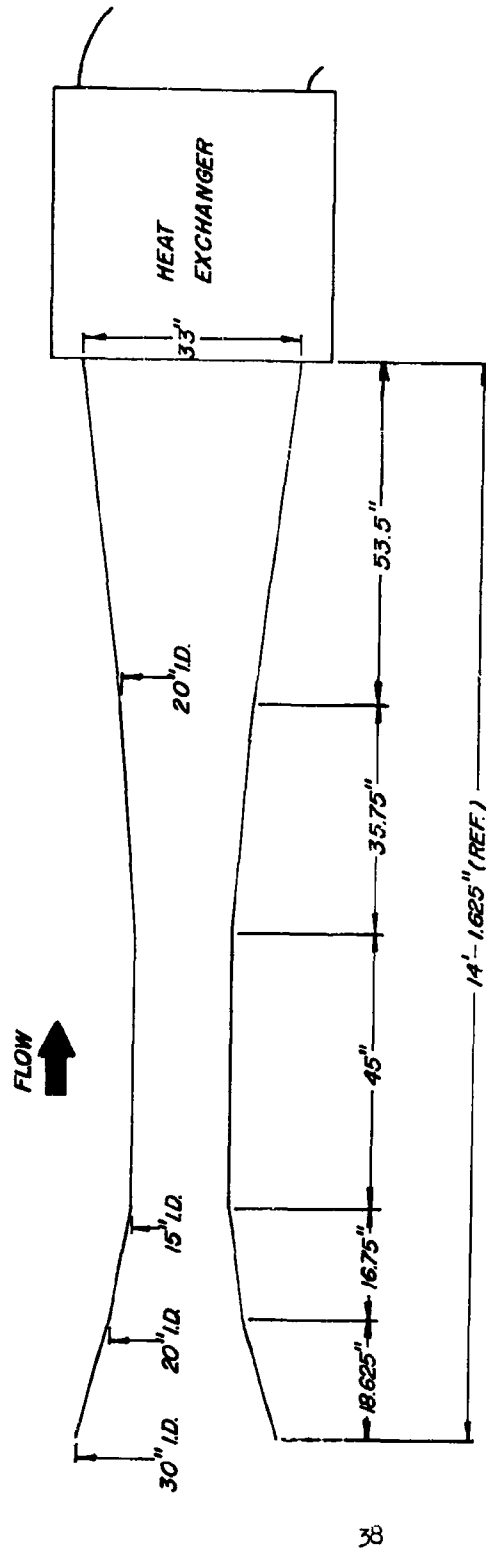
FIG. 9   DIFFUSER   CONFIGURATION

38
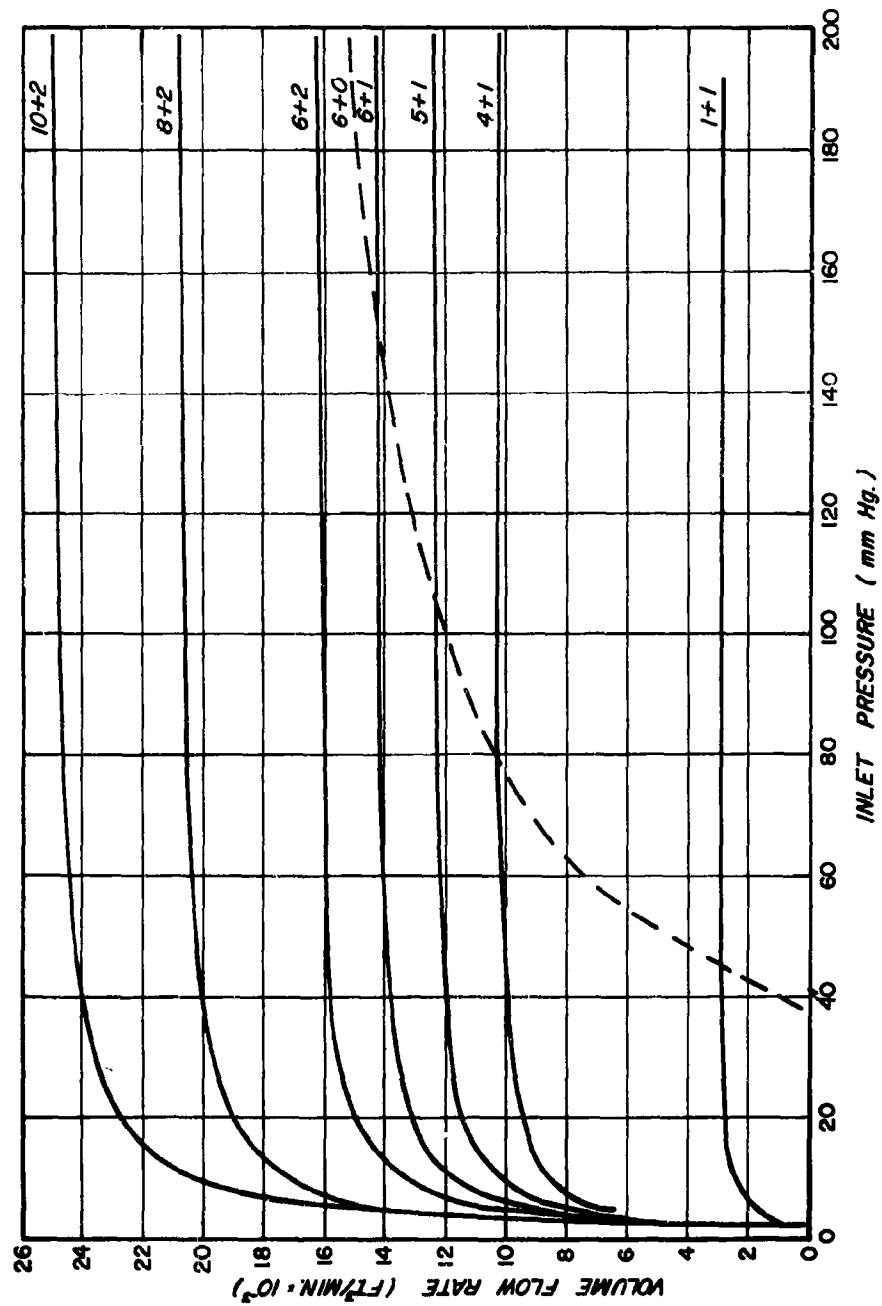
FIG. 10   VOLUME   FLOW   RATE vs.   INLET   PRESSURE   FOR   SEVERAL

VACUUM   PUMP   COMBINATIONS
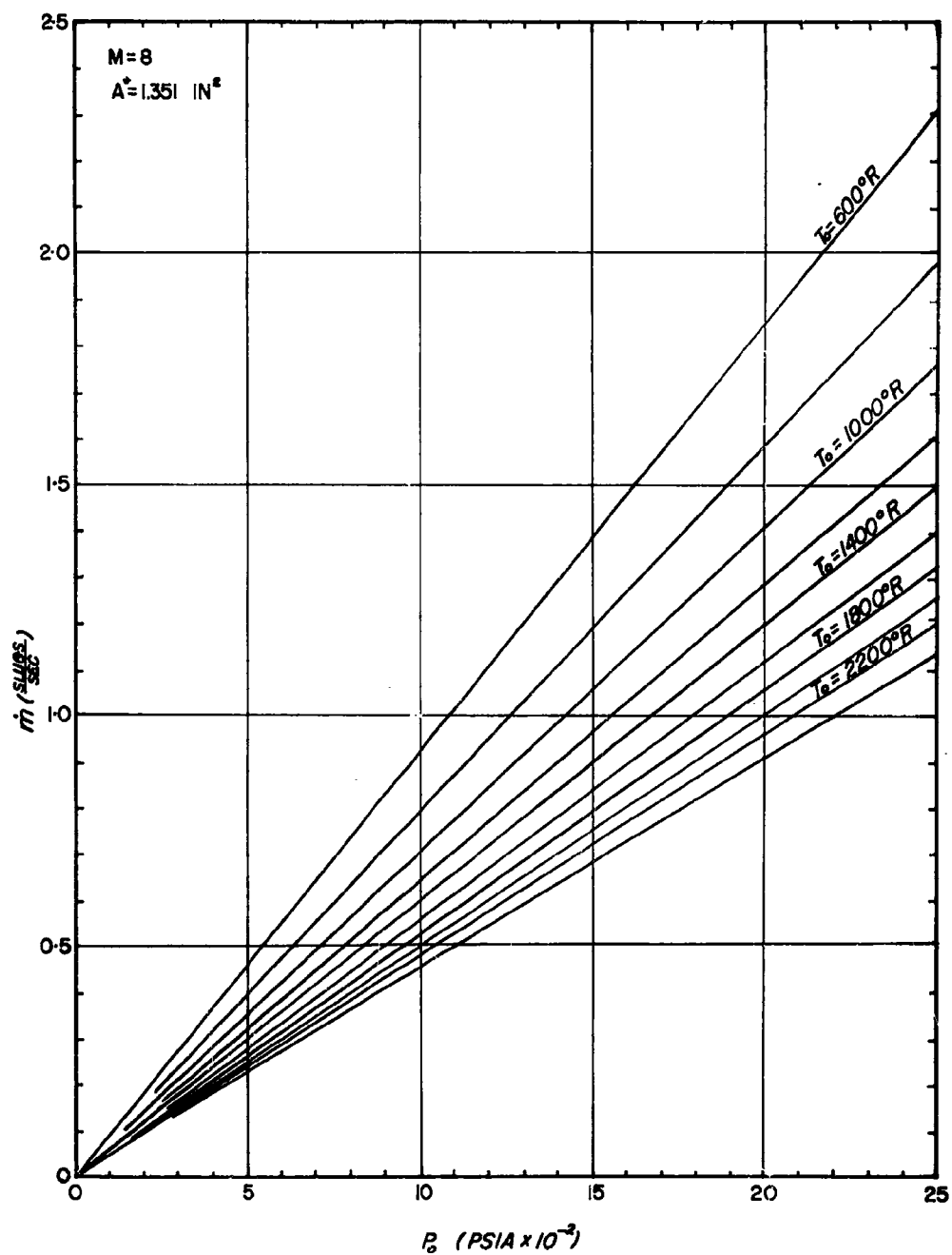
FIG. IIa   MASS   FLOW   RATE vs. STAGNATION   PRESSURE.
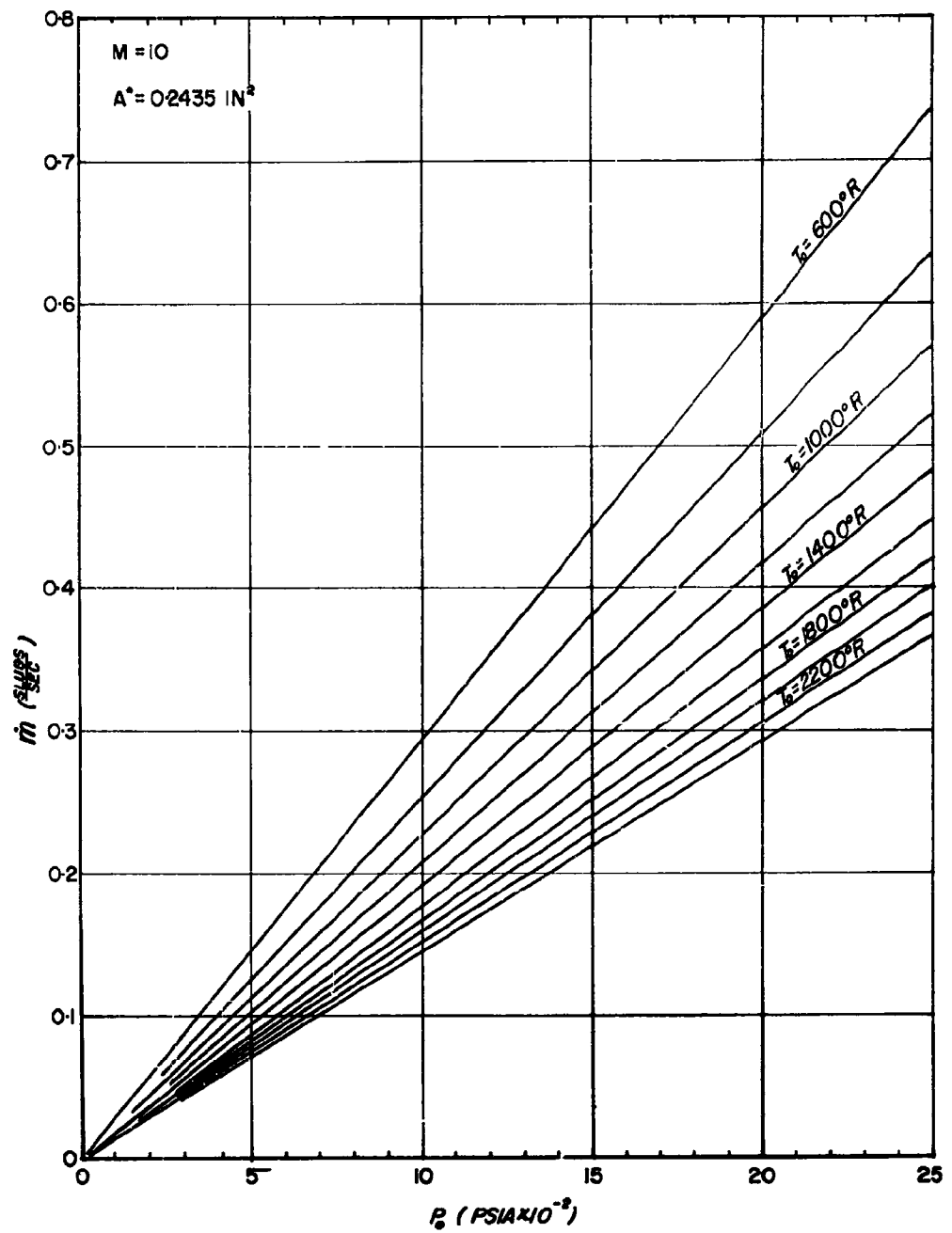
FIG. IIb   MASS   FLOW   RATE vs. STAGNATION   PRESSURE.
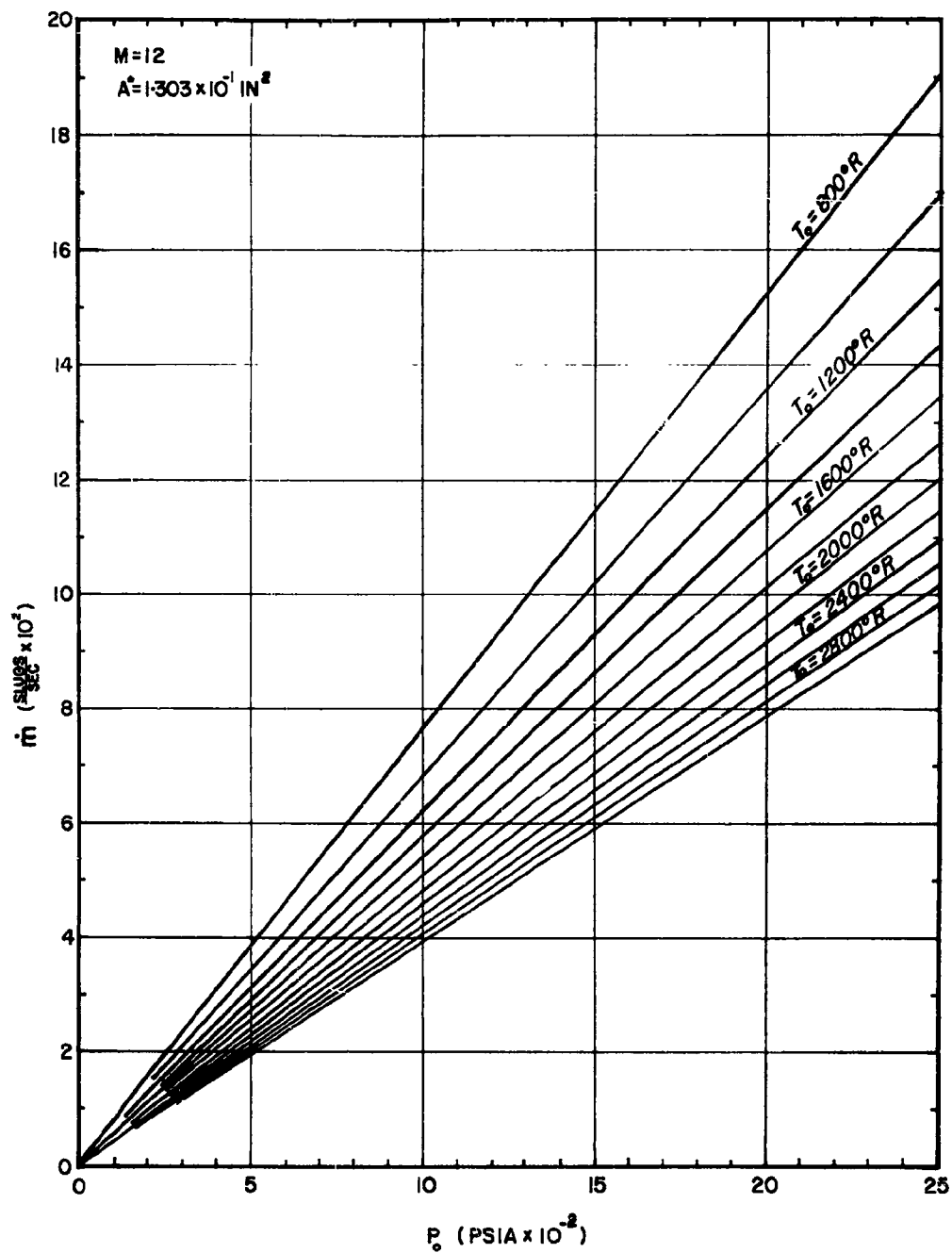
FIG. IIc   MASS FLOW RATE *vs.* STAGNATION PRESSURE.
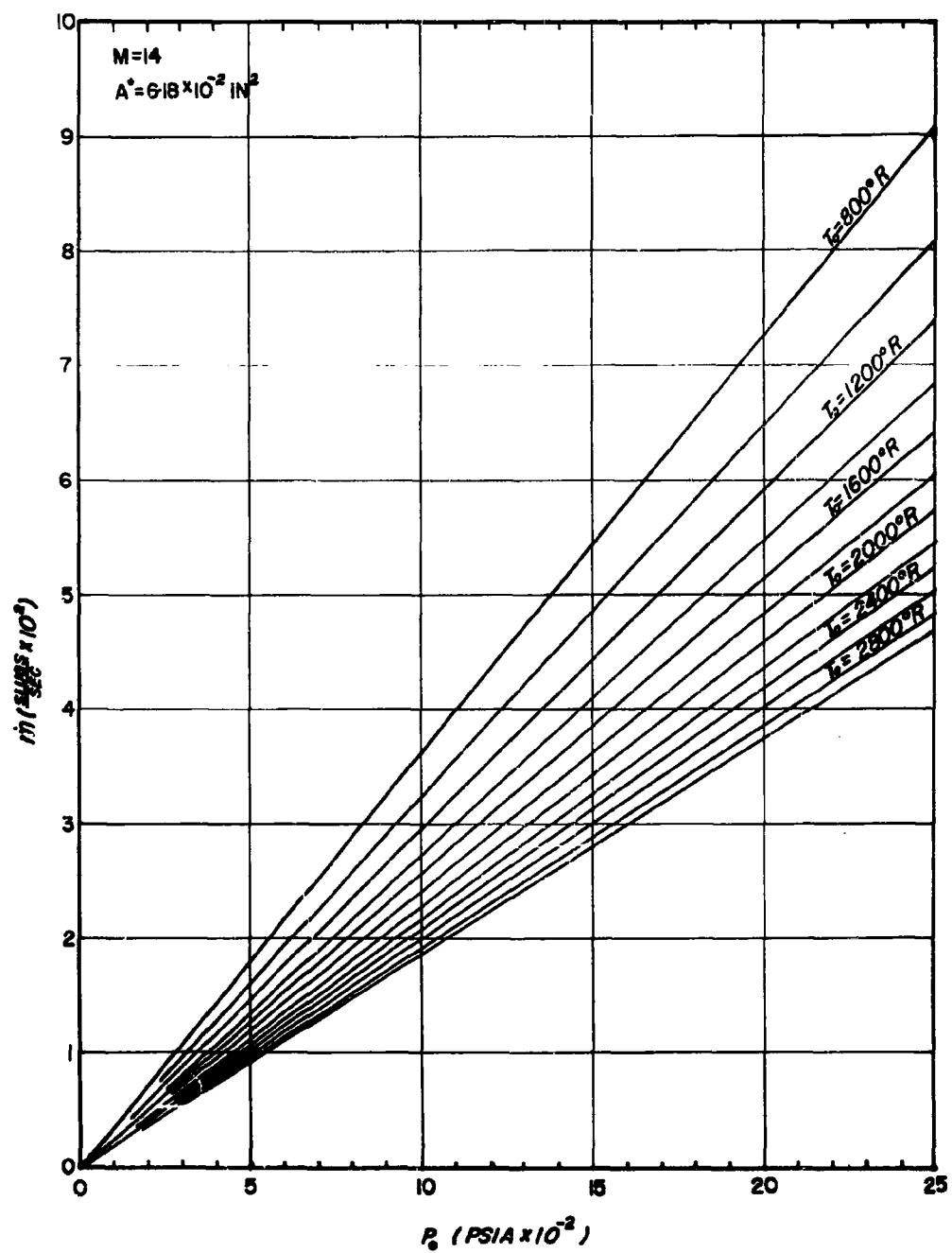
FIG. IId MASS FLOW RATE vs STAGNATION PRESSURE
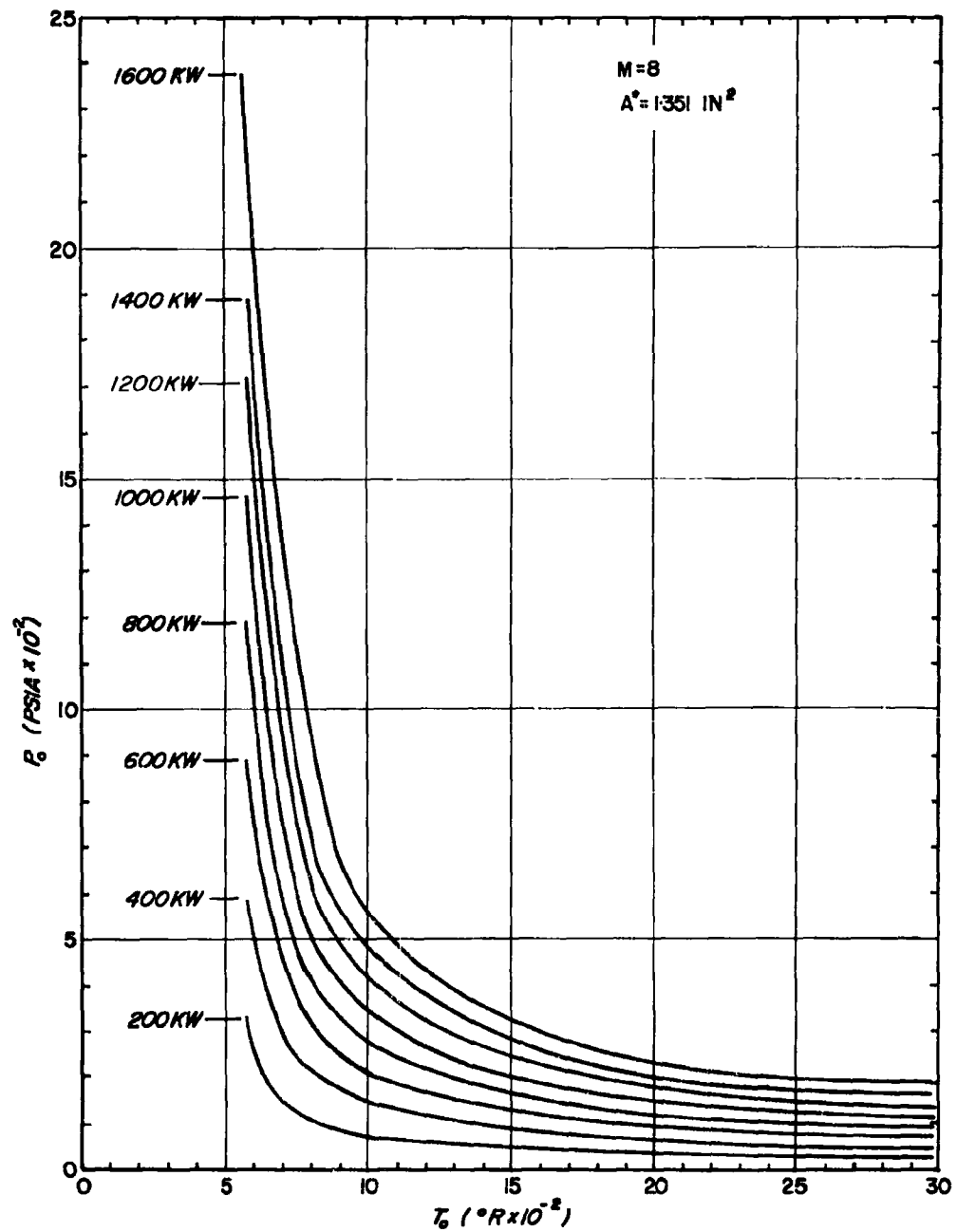
FIG. 12a STAGNATION PRESSURE *vs* STAGNATION TEMPERATURE
FOR CONSTANT POWER INPUT.
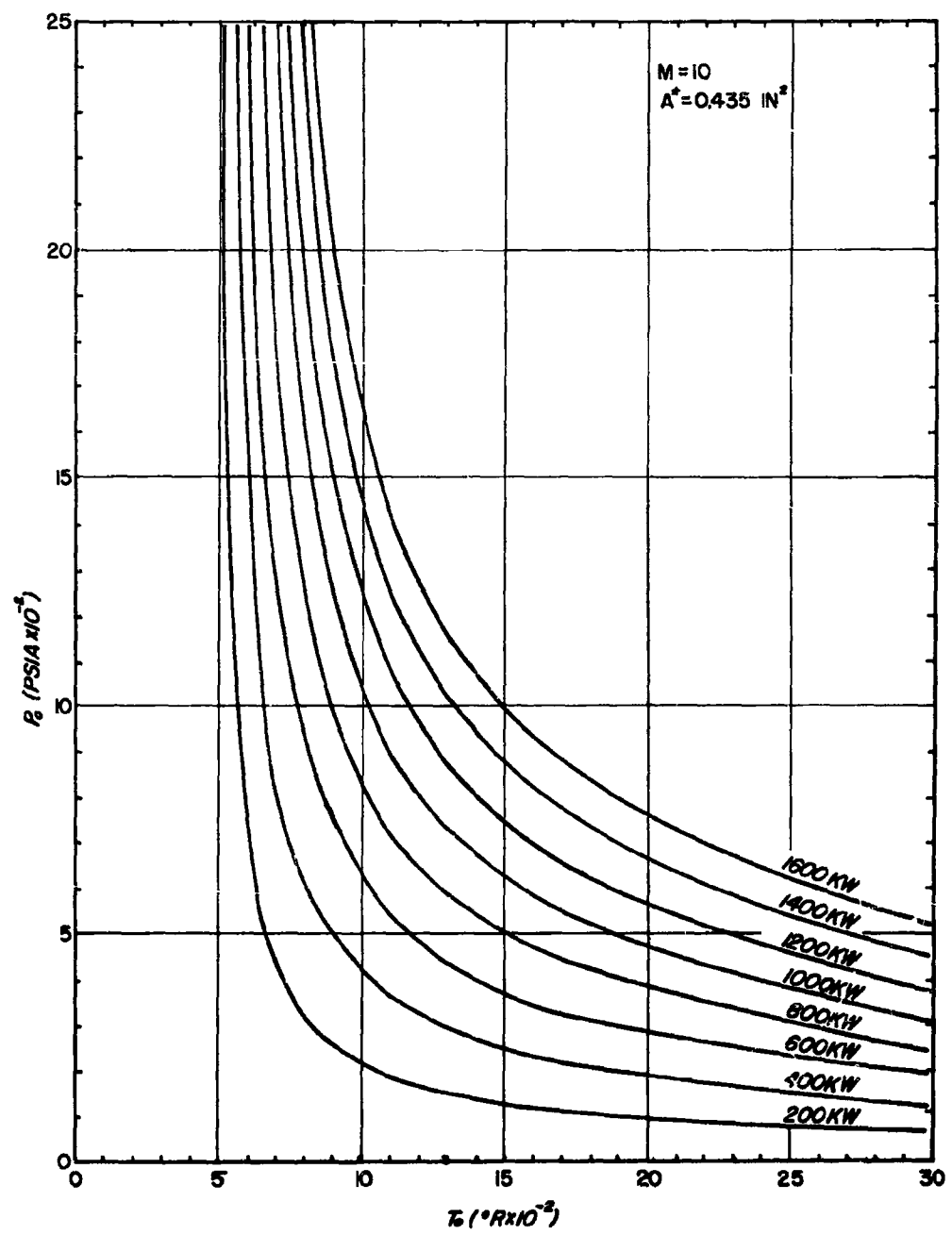
FIG. 12b STAGNATION PRESSURE vs. STAGNATION TEMPERATURE

45
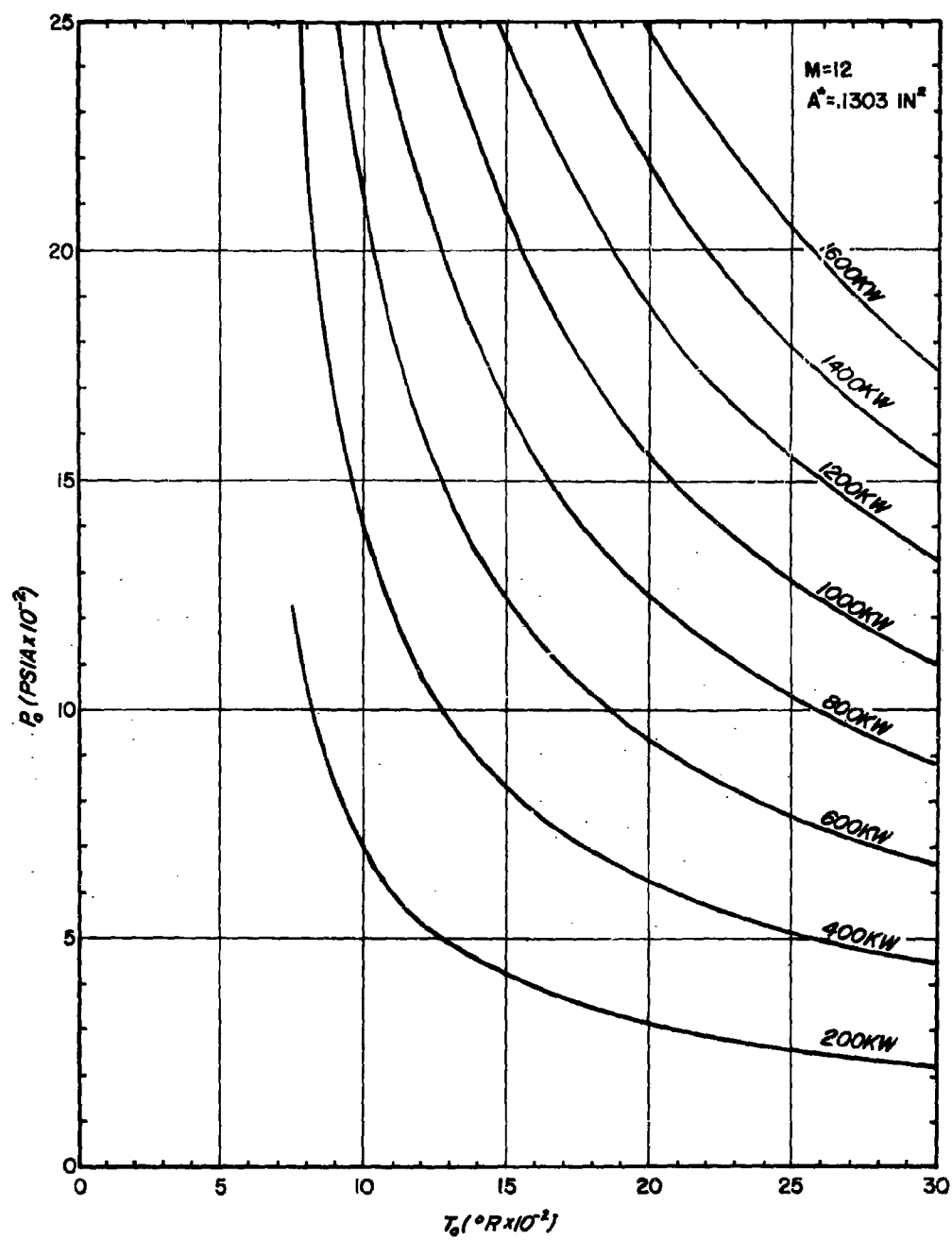
FIG. 12c STAGNATION PRESSURE vs. STAGNATION TEMPERATURE
FOR CONSTANT POWER INPUT.

46
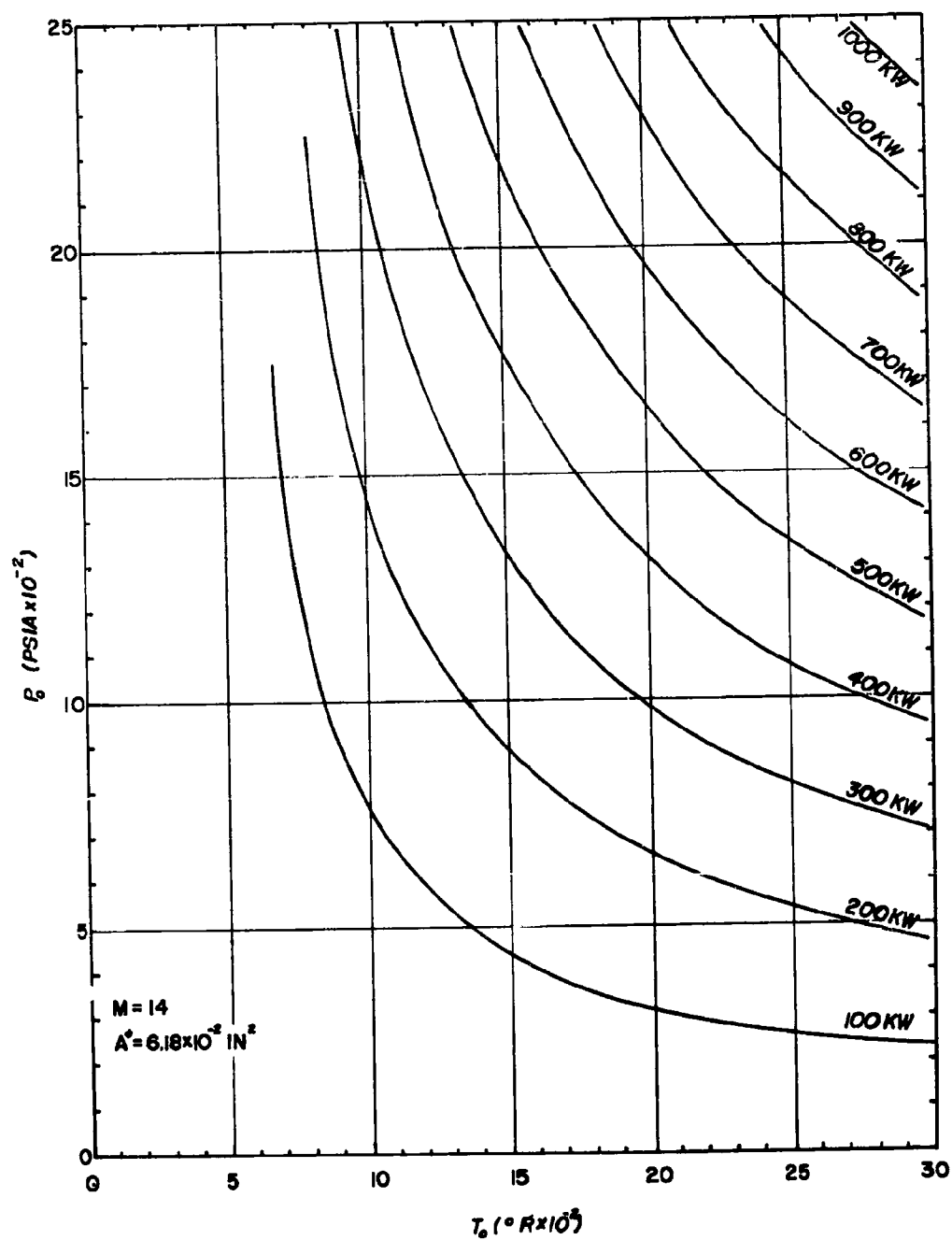
FIG. 12d STAGNATION PRESSURE vs. STAGNATION TEMPERATURE FOR CONSTANT POWER INPUT.

47
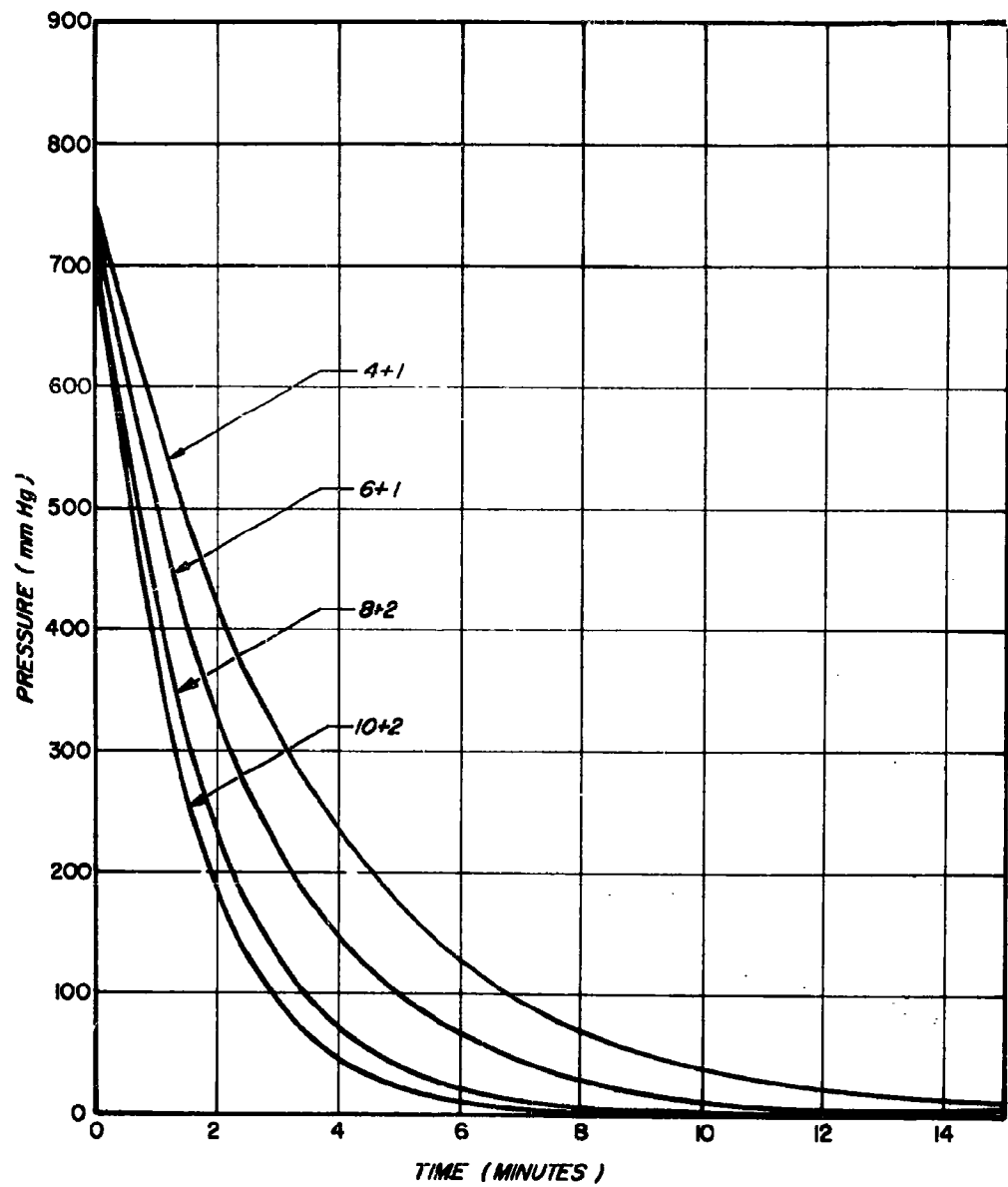
FIG. 13a EVACUATION RATE OF SPHERE FROM 1 ATM.
FOR FOUR VACUUM PUMP COMBINATIONS.

48
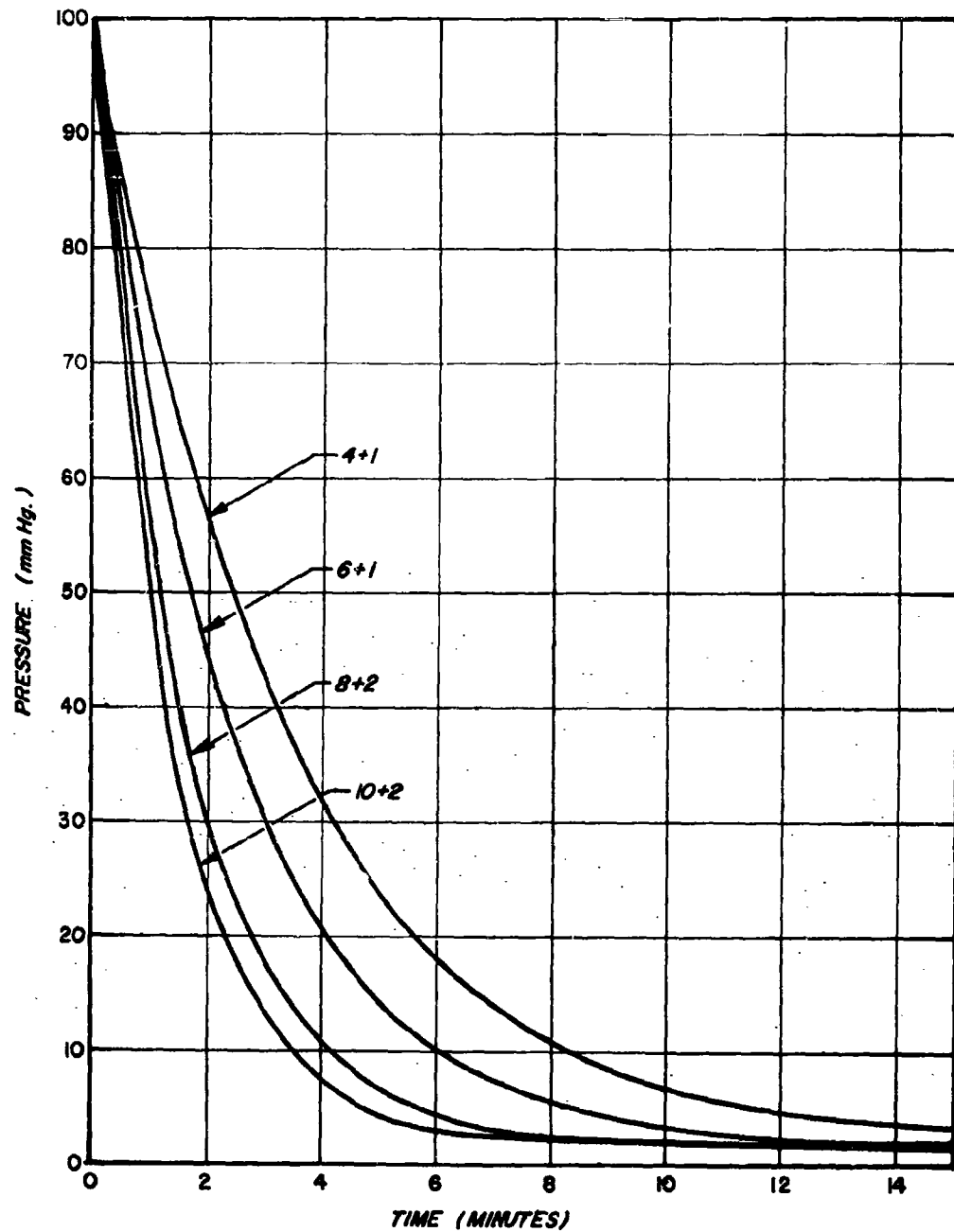
FIG. 13b EVACUATION RATE OF SPHERE FROM 100mm Hg.
FOR FOUR VACUUM PUMP COMBINATIONS.

49
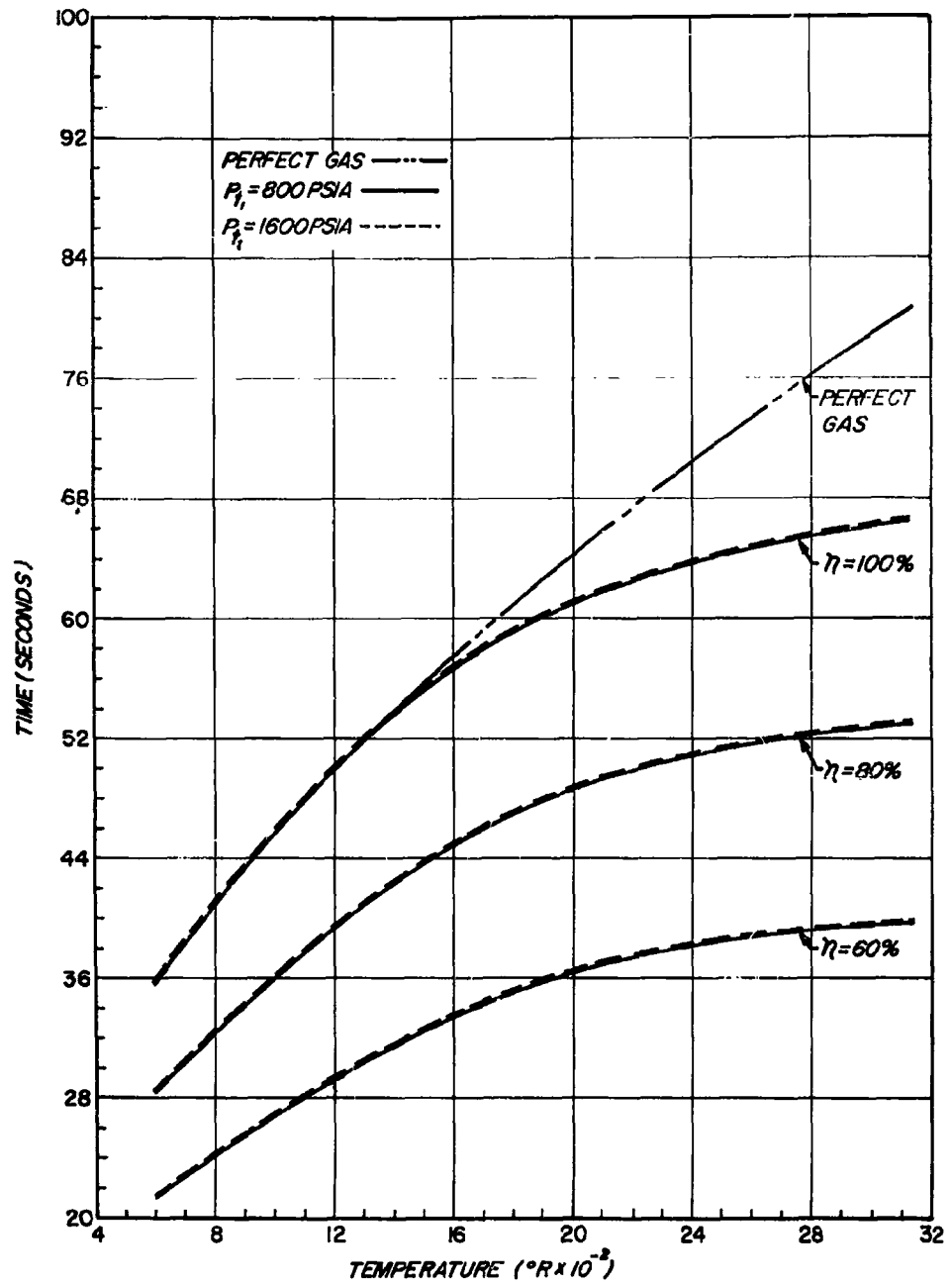
FIG. 14a RUN TIME vs. STAGNATION TEMPERATURE, M=8

50
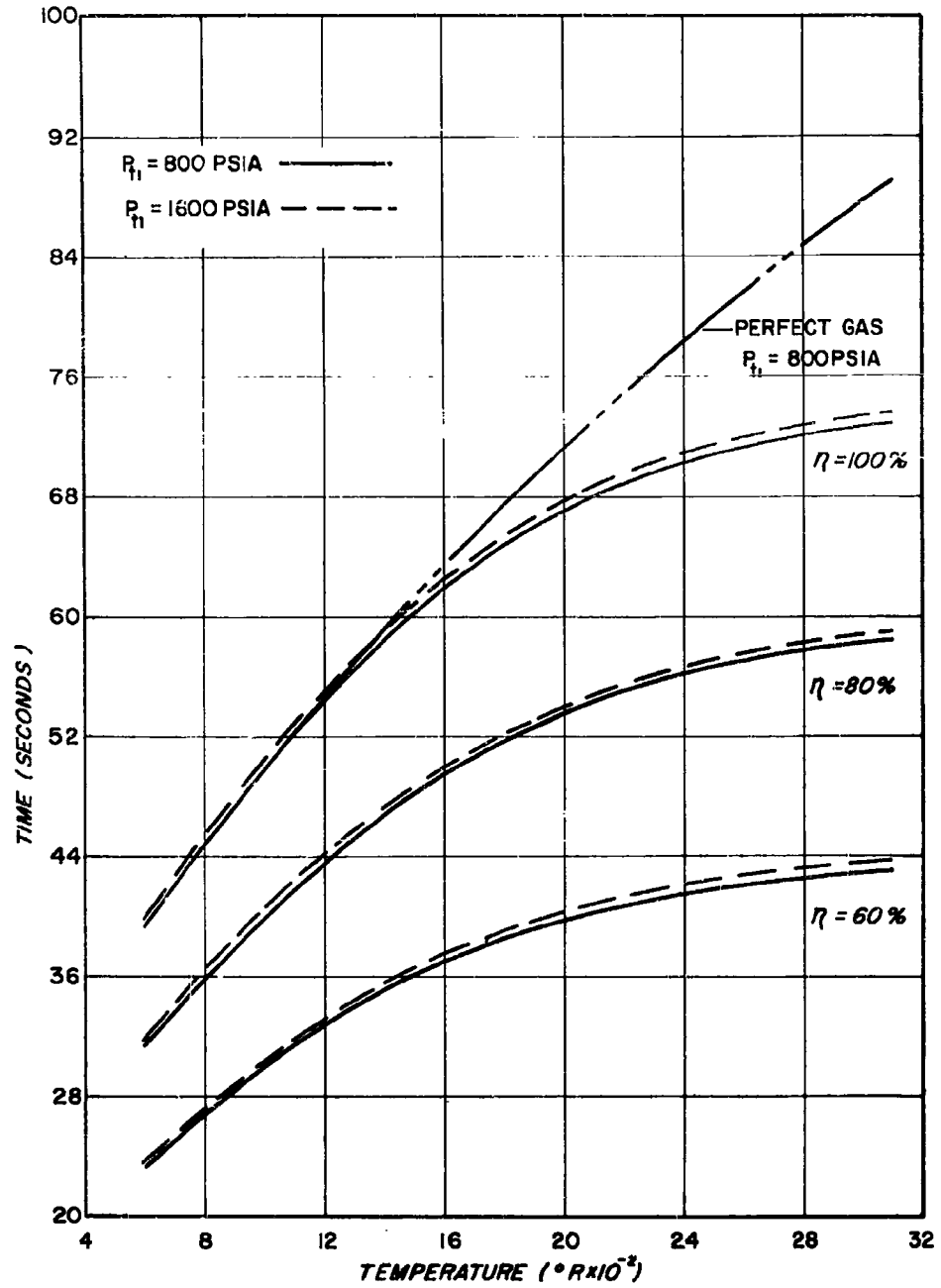
FIG. 14b RUN TIME vs. STAGNATION TEMPERATURE, M = 10

51
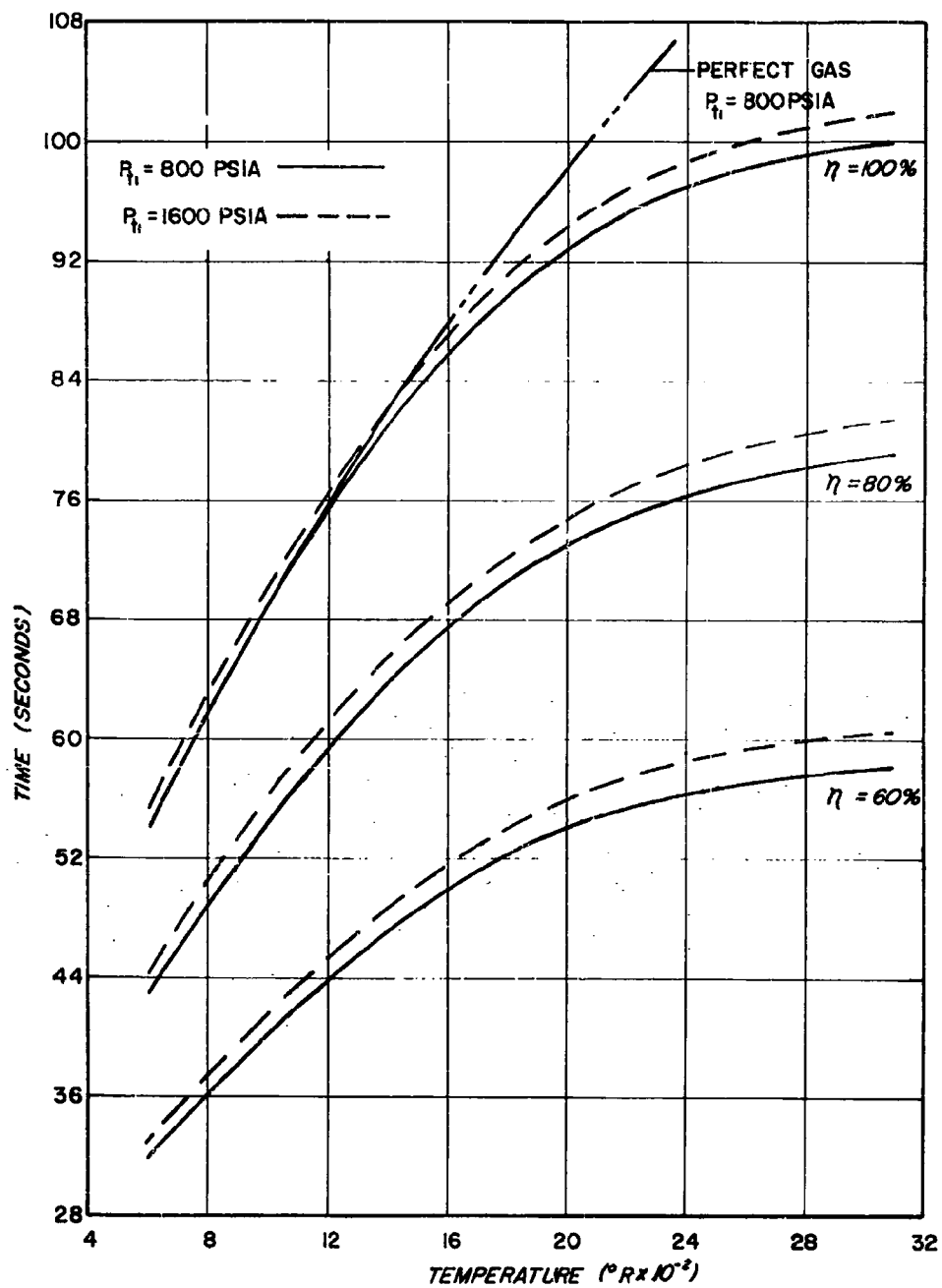
FIG. 14c RUN TIME *vs.* STAGNATION TEMPERATURE, M =12.
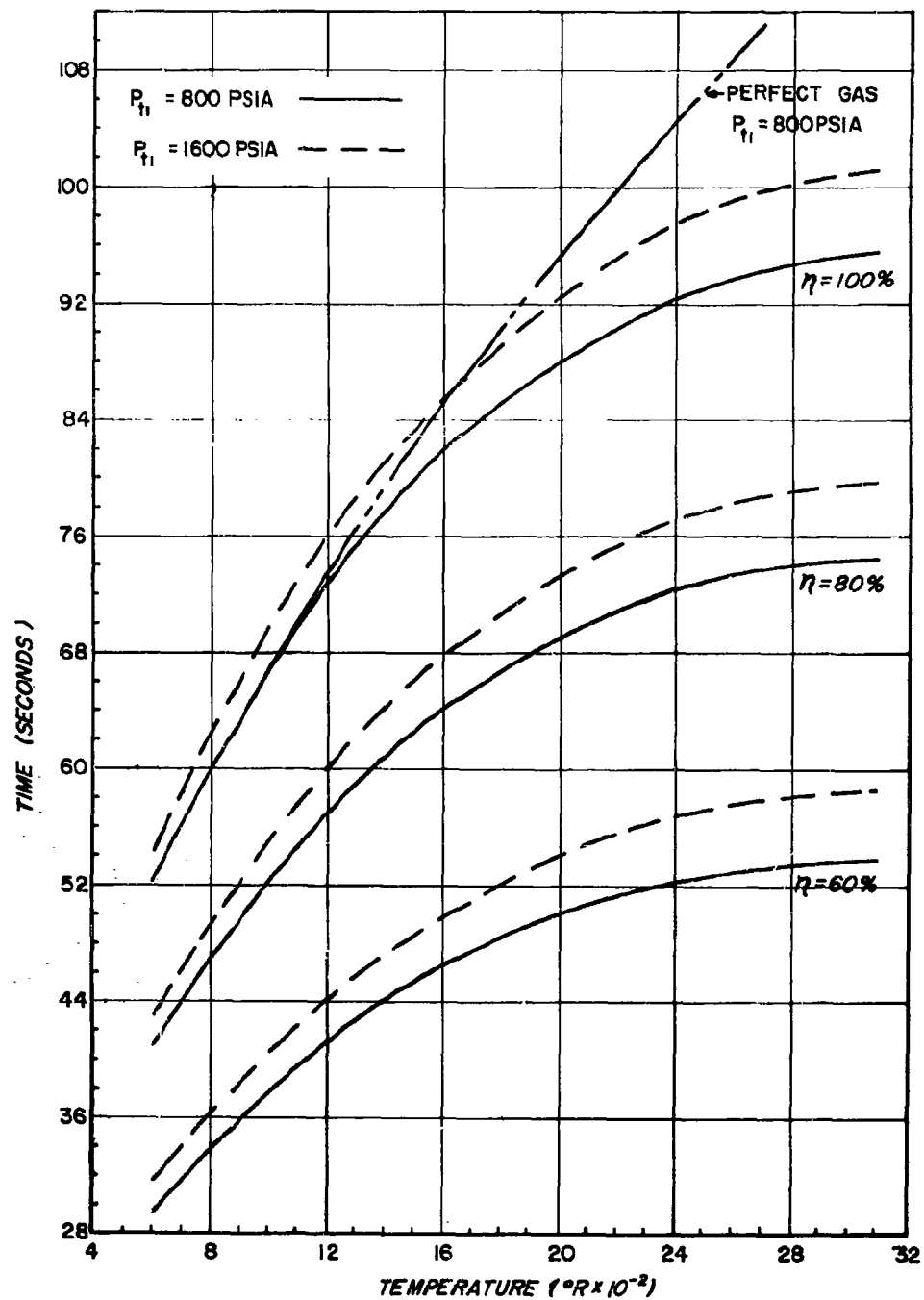
FIG.14d RUN TIME *vs.* STAGNATION TEMPERATURE, M = 14

53
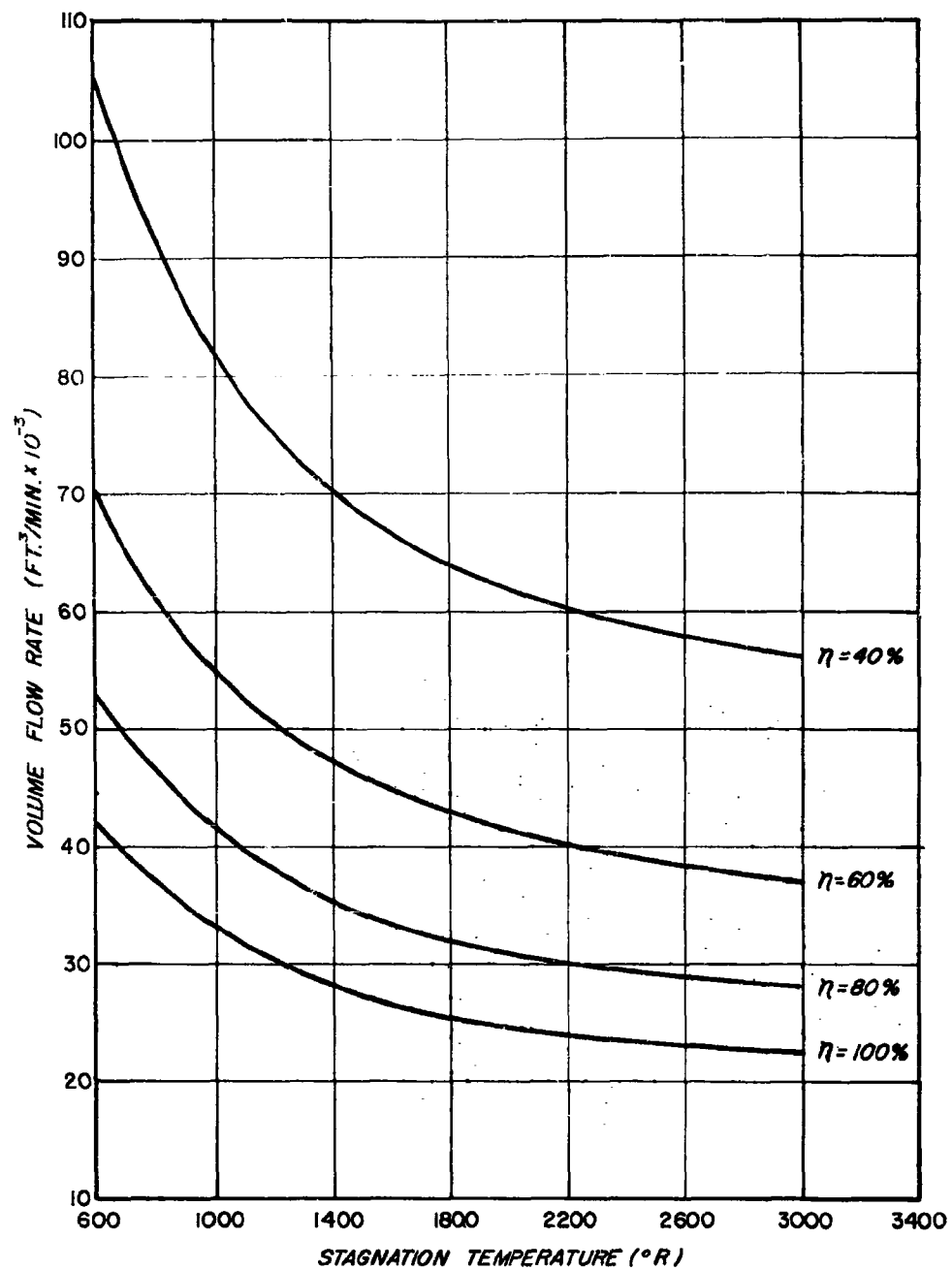
FIG. 15a   VOLUME   FLOW   RATE   FROM   HEAT   EXCHANGER
AT   MACH   8.

54
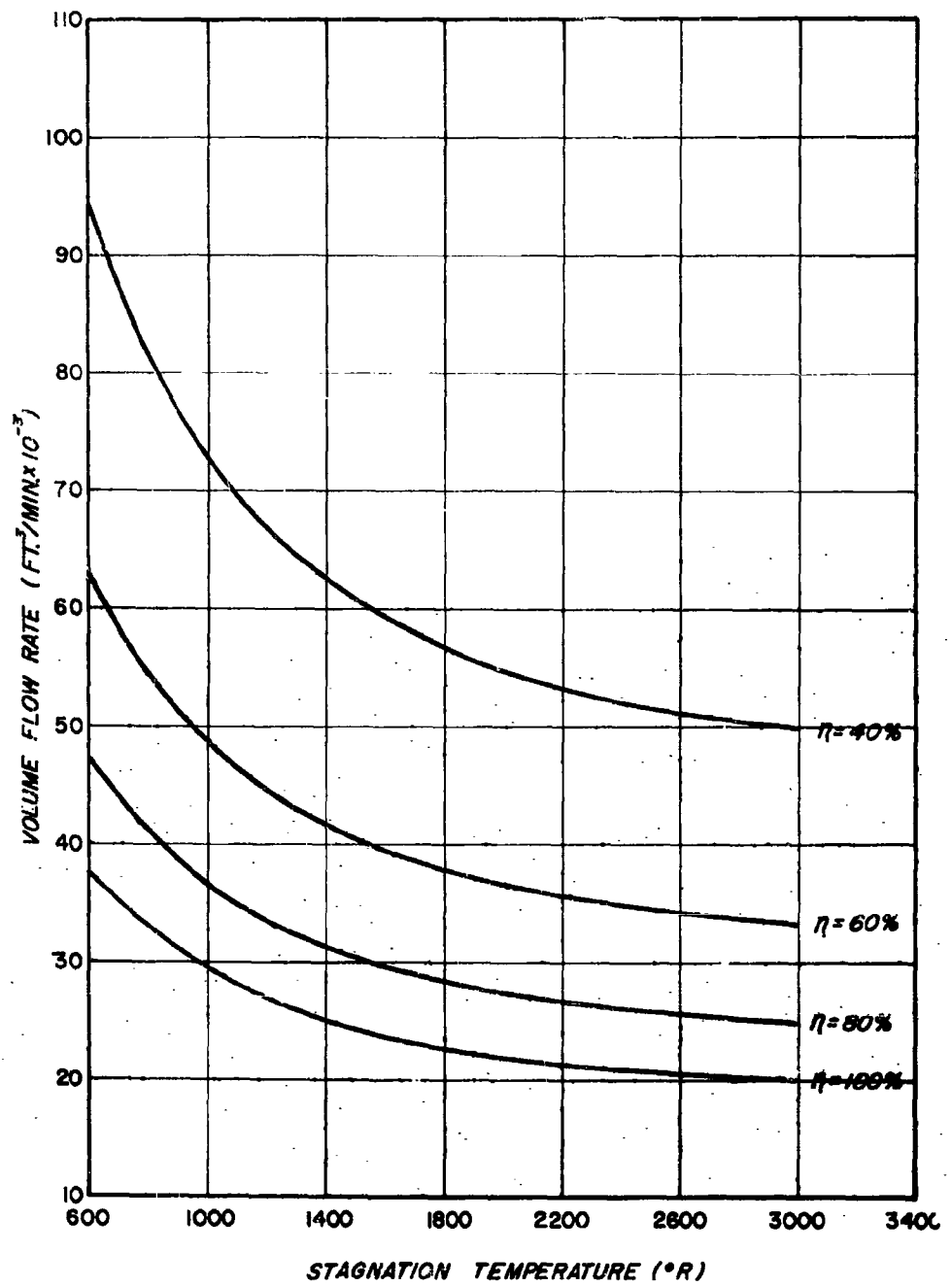
FIG. 15b VOLUME FLOW RATE FROM HEAT EXCHANGER
AT MACH 10.

55
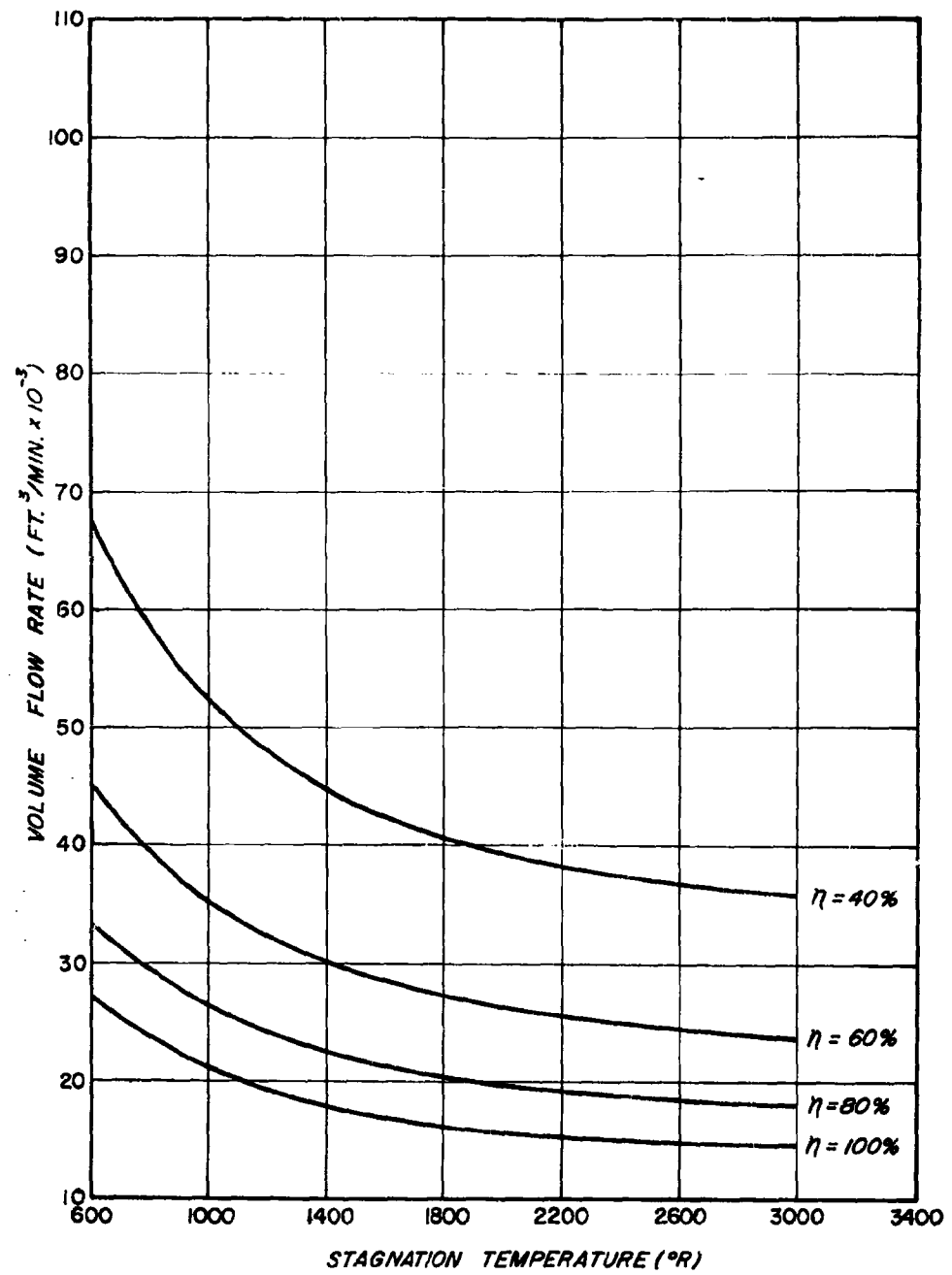
FIG. 15c VOLUME FLOW RATE FROM HEAT EXCHANGER
AT MACH 12

56
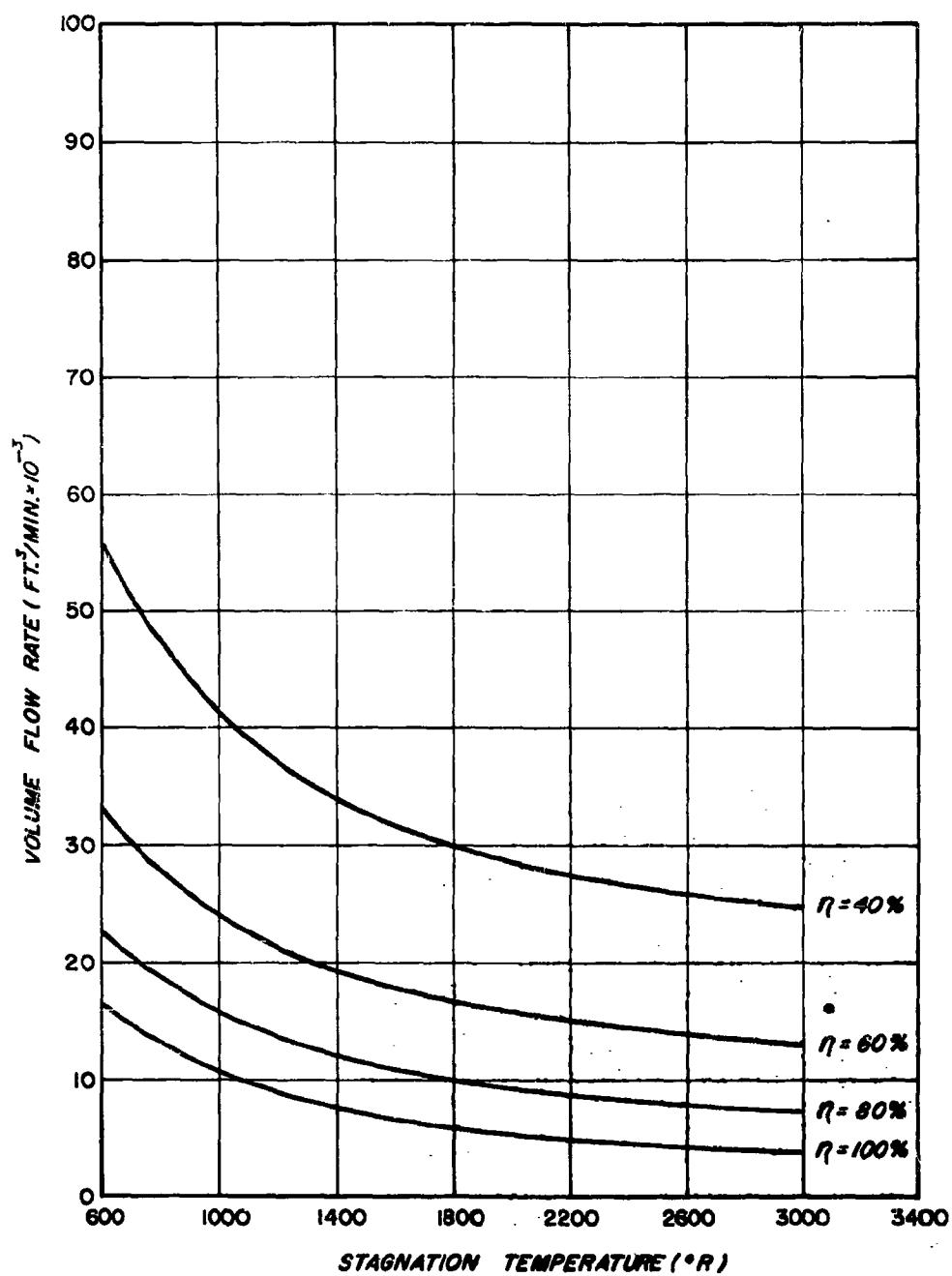
FIG. 15d VOLUME FLOW RATE FROM HEAT EXCHANGER
AT MACH 14.
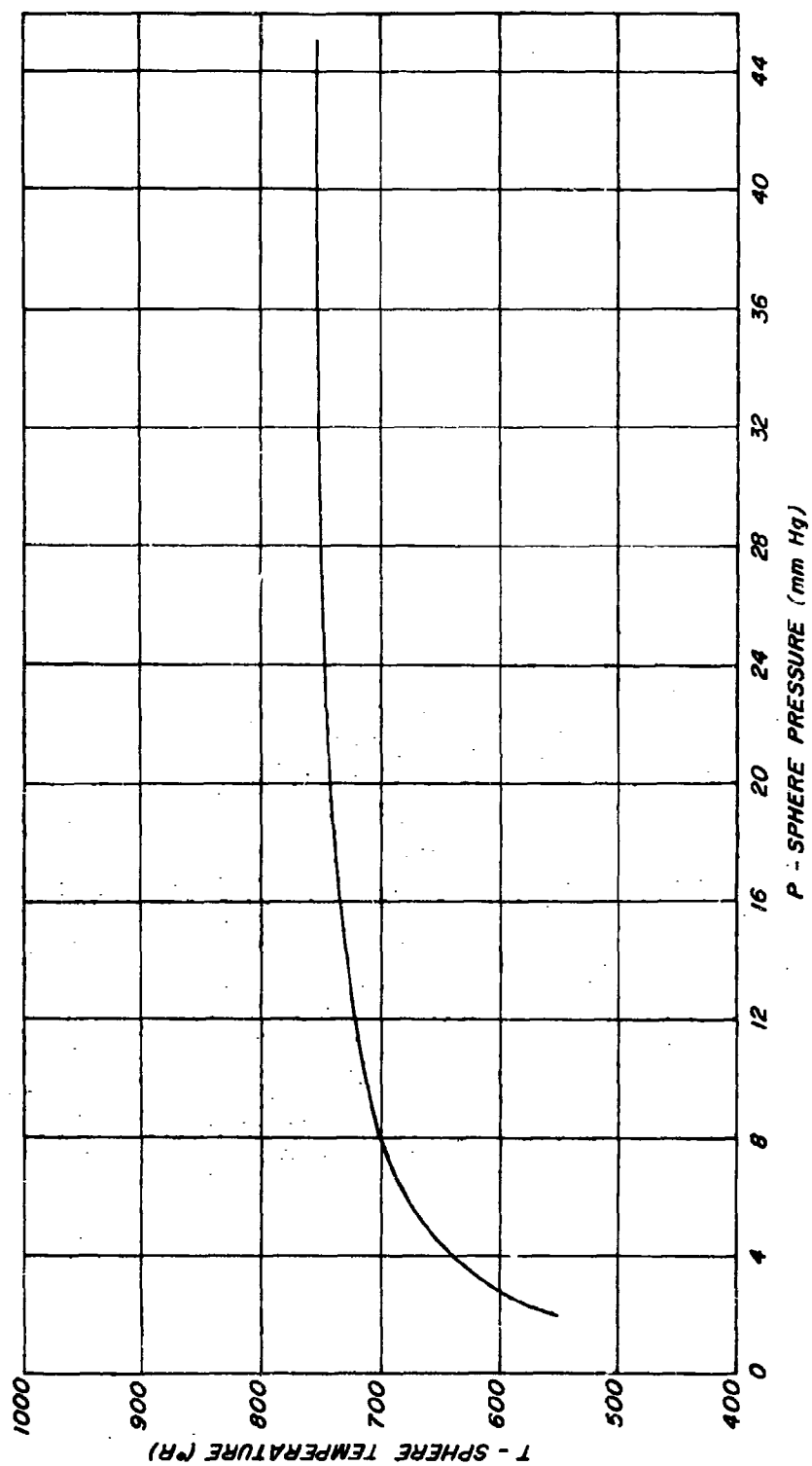
FIG. 16 SPHERE TEMPERATURE DURING FILLING

58
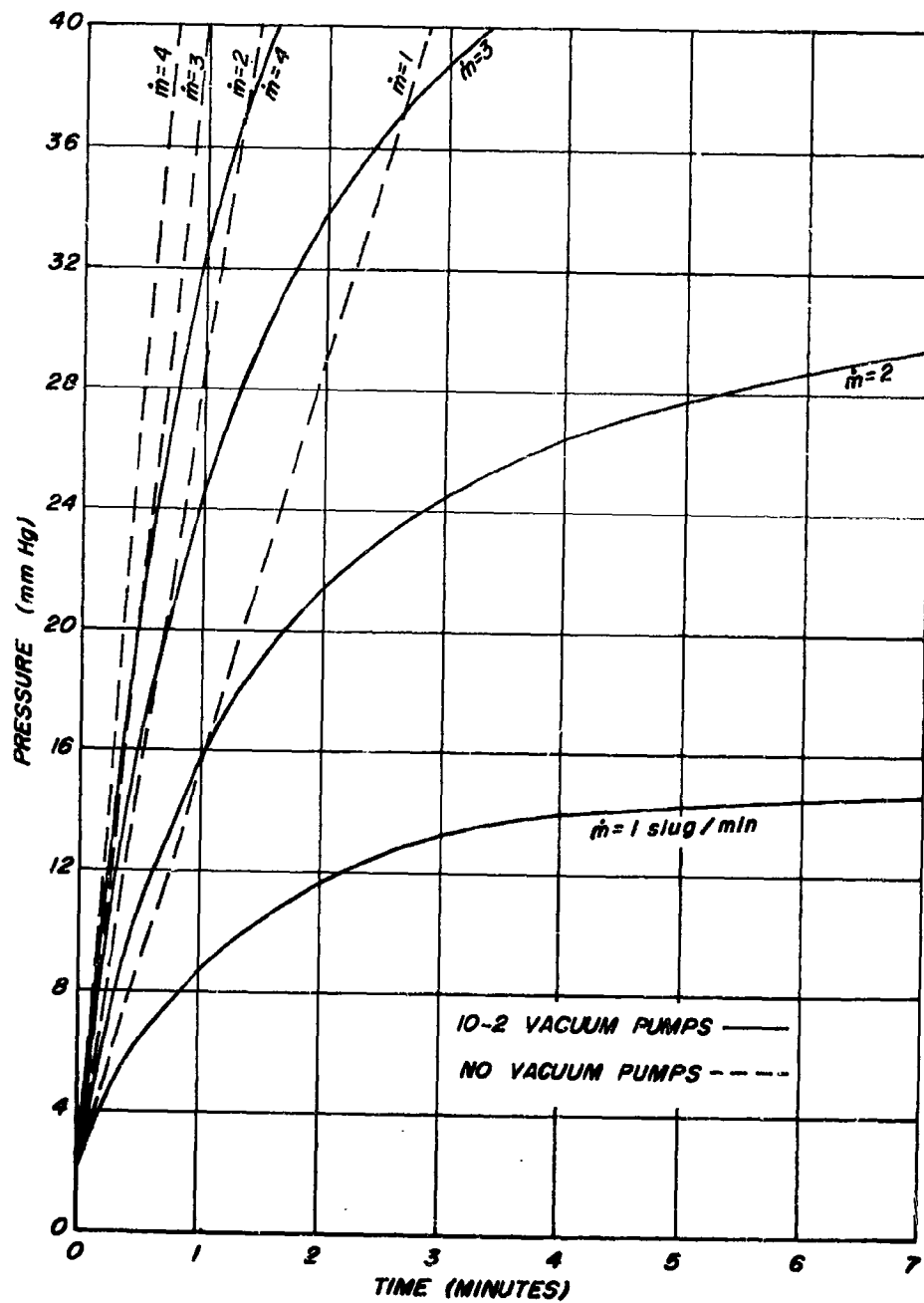
FIG.17 SPHERE PRESSURE vs TIME DURING
ADIABATIC FILLING WITH AND WITHOUT
VACUUM PUMPS

59
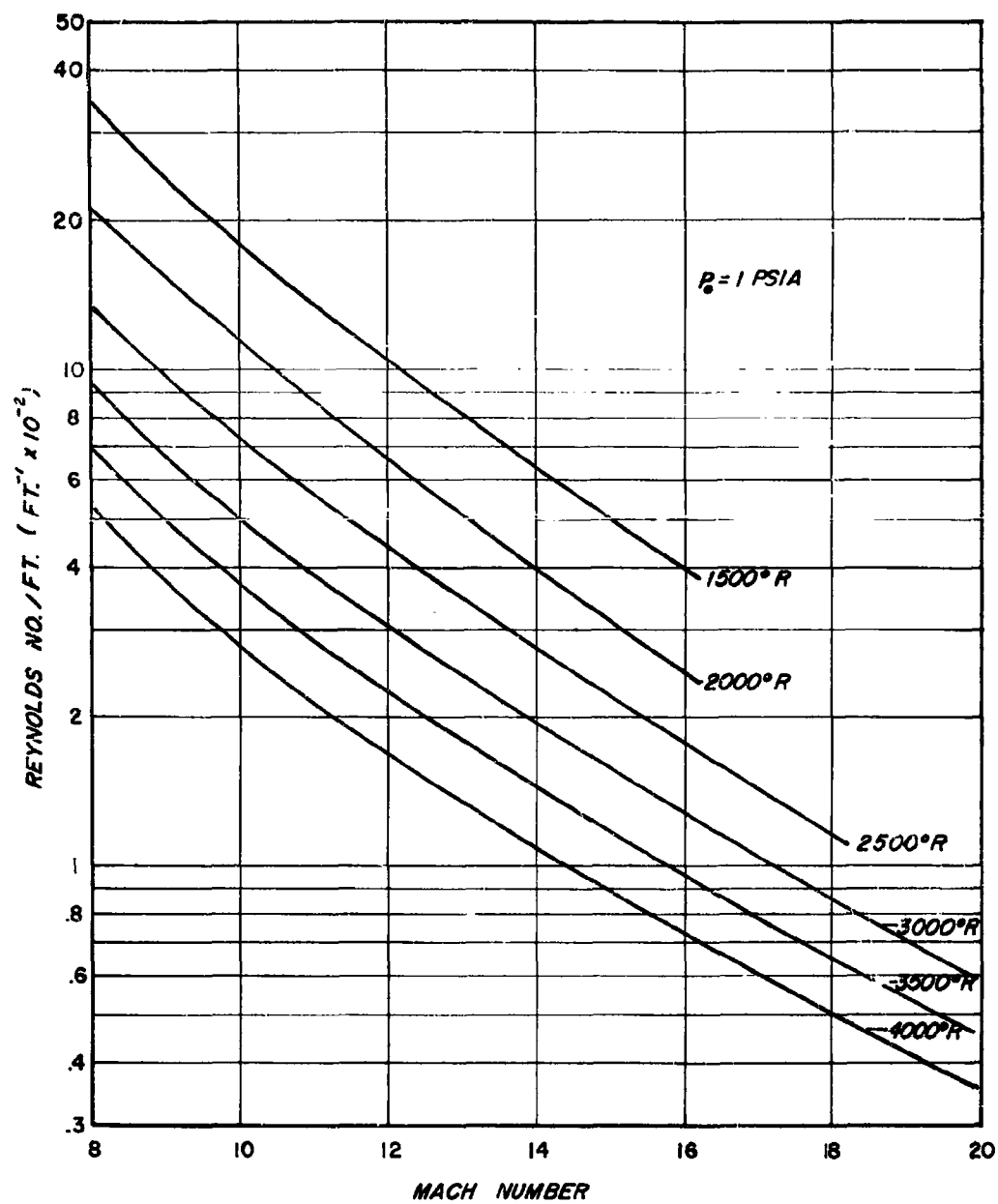
FIG.18   REYNOLDS NUMBER/ FOOT *vs.* MACH NUMBER.
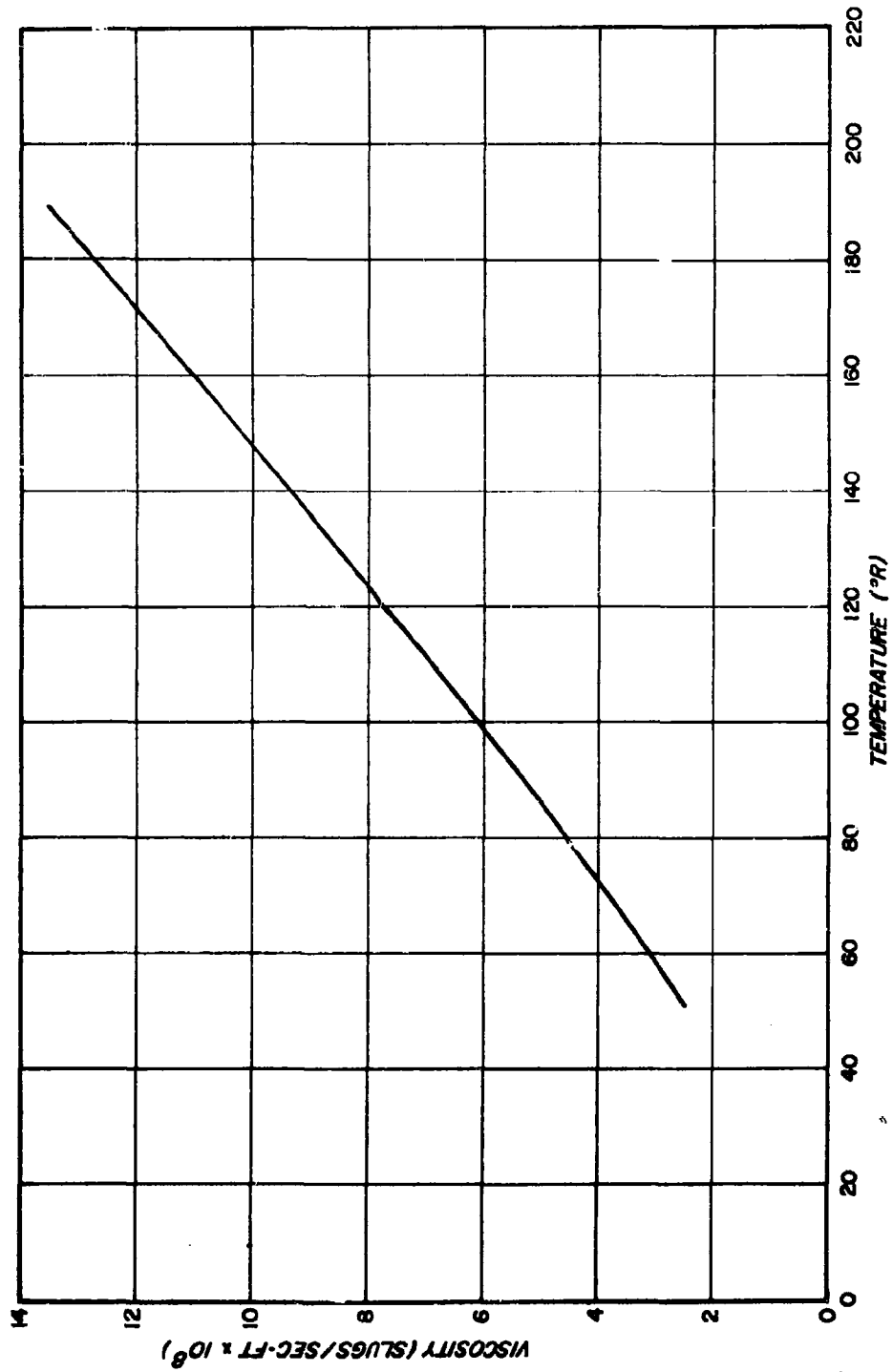
FIG. 19 **VISCOSITY OF AIR AT LOW TEMPERATURES** REF: BROMLEY-WILKE DATA,
UNIVERSITY OF CALIFORNIA, T.R. HE-150-157.

61
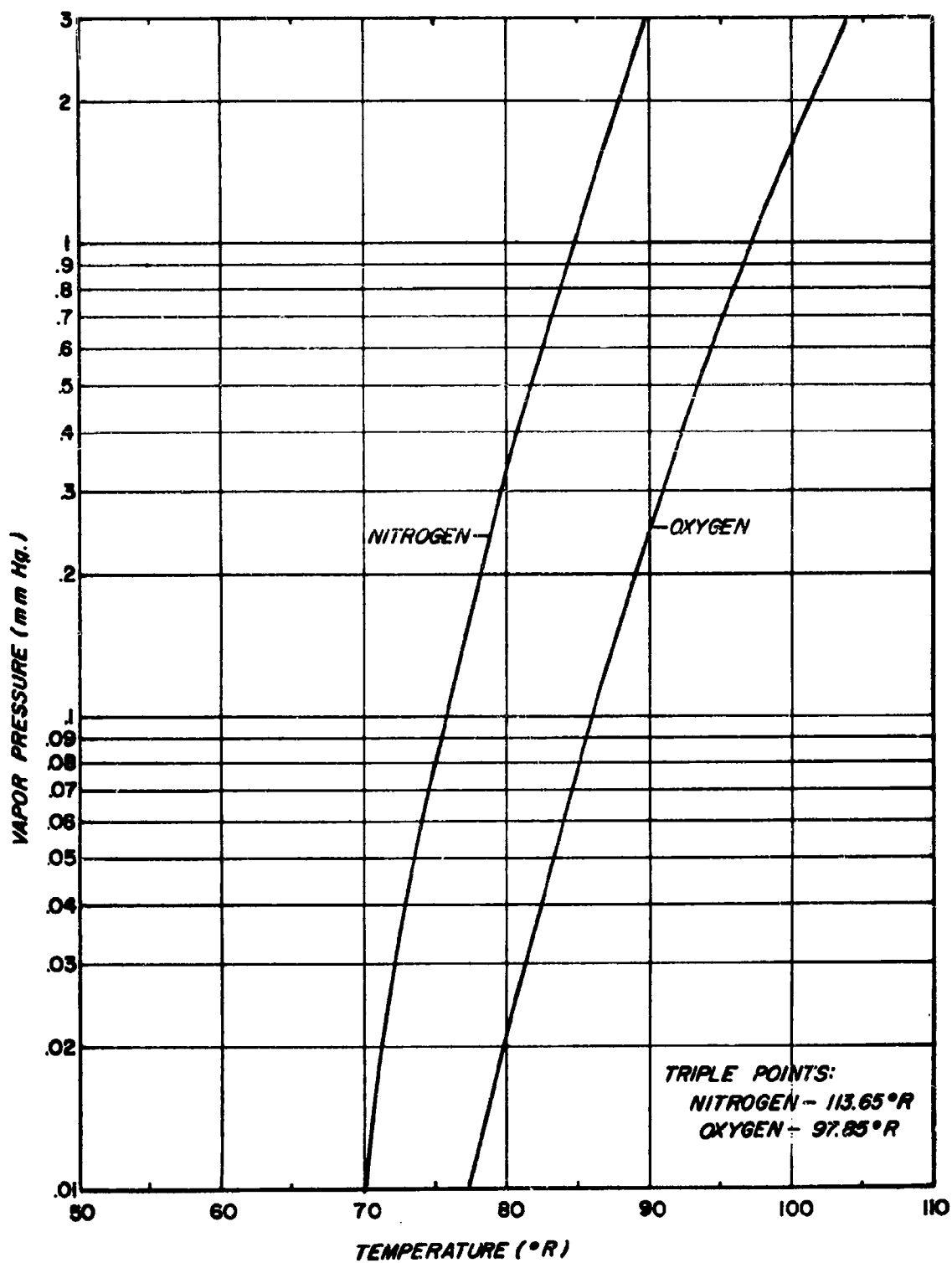
FIG. 20   SATURATION   CONDITIONS   FOR   OXYGEN   AND
NITROGEN   PRESSURE   vs.   TEMPERATURE.

Aeronautical Research Laboratories, Wright-
Patterson AFB, Ohio. DESIGN PERFORMANCE AND
OPERATIONAL CHARACTERISTICS OF THE ARL
TWENTY-INCH HYPERSONIC WIND TUNNEL by G.M.
Gregorek, J. D. Lee. August 1962. 70 p. incl.
illus. (Project 7065; Task 7065-01) (Contract
AF 33(616)-7451) (ARL 62-392)

Unclassified Report

The ARL Twenty-Inch Hypersonic Wind Tunnel
is a blow-down type, free-jet installation
that can deliver air at Mach numbers from 8
to 14 at free stream Reynolds numbers of 10^7
per foot to 0.3×10^6 per foot.

The report discusses some of the initial
considerations influencing the configuration

(over)

of the facility and presents theoretical per-
formance estimates of its aerodynamic com-
ponents; i.e., the electric resistance heater,
the nozzle, the diffuser and the vacuum sup-
ply. Operational characteristics such as
mass flow rates, electric power requirements,
sphere evacuation rates, etc., are also
treated for the full range of Mach numbers.
Brief discussions on the extension of test-
ing time and on the character of the de-
livered air flow are included.

Mechanical design, control systems, and
wind tunnel instrumentation of the facility
are not considered.